PRACTICAL THERMODYNAMICS

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PRACTICAL THERMODYNAMICS

A TREATISE ON THE THEORY AND DESIGN OF HEAT ENGINES, REFRIGERATION MACHINERY, AND OTHER POWER-PLANT APPARATUS

BY

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PREFACE

In writing this volume the author has endeavored to present the natural laws and physical principles which underlie the action of thermodynamic apparatus in such a manner as to enable the student not only to comprehend the principles upon which the apparatus depends for its operation, but also to assist him to correctly design such an apparatus, to operate it, and to judge of its value.

The book is intended primarily as a text-book for the use of junior and senior classes in mechanical and electrical engineering. It therefore attempts to present the subject of thermodynamics from the physical rather than from the mathematical standpoint. While the treatment of the subject is rigorous, a minimum of higher mathematics is used and the methods employed are those which appeal to common sense rather than to a knowledge of calculus. The writer believes that the higher mathematics are to be regarded merely as a set of tools by the use of which the engineer may accomplish certain results. Hence the methods employed in developing the mathematical part of the work depend upon the objects to be accomplished, and are the simplest and most effective possible. They are usually methods which lay stress on the physical phenomena rather than those which appeal to the accomplished mathematician.

The writer has not hesitated to present new methods whenever these methods seem to be simpler and better than the older ones. Much of the difficulty of teaching thermodynamics arises out of a misunderstanding on the part of the student, of the phraseology usually employed, and from attempts to introduce the abstractions of the ancient philosophers into the concretions of modern science. Accordingly, no chapters are devoted to the first or second laws of thermodynamics and no puzzling or troublesome analogies are offered for entropy. Many definitions differ radically from those offered in other text-books, but the changes made are considered advisable in order to make the presentation of the subject more logical or more simple.

It does not seem to the author that it is any longer necessary or desirable to give credit to the originators of methods of thermodynamic

investigation, or to the discoverers of physical truths long known, in a work of this kind. Accordingly, he has refrained from distracting the student's attention by continual reference to names which are strange to him and which, while they are of historic interest, should have no place in a text-book. While this book contains very little that is new, and is principally a presentation of truths long known, yet authorities are quoted but seldom. The effect of such references and quotations upon the student's mind is usually to divert his thought from the principles to be considered, and they are therefore omitted unless some unusally good reason dictates their insertion.

The author has separated all reference to the temperature-entropy diagram from the body of the text and placed it it a separate chapter (the 25th) in order that it may be used or not as the judgment of the teacher shall dictate. The material in the 25th chapter is so arranged that it exactly parallels the remainder of the book, and the temperature-entropy analysis of any type of thermodynamic machine may be found in its proper order. It is the author's opinion that the temperature-entropy diagram is not illuminating to the average student, who is continually seeking for some physical analogy for entropy and who is continually plunging himself into difficulties by seeking to carry his analogy further than the truth will warrant. The use of the entropy function in developing the theory of the adiabatic expansion of vapors, on the other hand, involves no difficulties whatever, and the author has therefore presented this matter in the body of the text.

The writer has attempted at all times to bring most of the work well within the comprehension of the average technical student. In some places it will seem as if he had made his treatment of the work absurdly simple, as, for instance, in the description of the steam engine. He believes, however, that many of the difficulties encountered by students arise from a misunderstanding of facts which seem to the teacher to be perfectly obvious and which the teacher mistakenly believes that the student thoroughly understands. Another difficulty often encountered in teaching thermodynamics arises from the fact that an inadequate preparation precedes the study of the phenomena of heat engines and other thermodynamic machinery. The author has therefore endeavored to present in the first seven chapters of the book the fundamental physical principles upon which a further study of the subject must depend, in such a thorough and simple manner that no trouble need subsequently arise from a misunderstanding of these fundamentals.

In order that the book shall be available to the largest possible number of classes, the author has included many items, which, while they are of great interest, can be profitably taught only to advanced students or to classes in which the amount of time available for the subject is greater

than that usually taken. These items have been printed in smaller type than the remainder of the text, and the teacher can omit any of them that he may see fit, without omitting any of the essential parts of the work.

The problems have been carefully chosen to illustrate the principles treated in the text. The author believes that a student has a true knowledge of his subject, not when he can make a recitation of the substance of a statement in a text-book, but when his knowledge can be applied to the solution of problems. The problems have been so arranged as to advance the student one step at a time in his work, only one new element being introduced in each problem. By concentrating the student's attention upon the new element, the problems are made more easy of solution and are more valuable from an instructional standpoint. The problems are not intended as examination questions to show the student's grasp of his subject, but are rather intended to be suggestive to him and to assist him in the comprehension of the text. The answers accompany the problems in each case.

In working the problem it will in many cases be necessary to make use of a steam table. The reader should therefore procure such a table. Either Peabody's tables, published by John Wiley & Sons, or Marks and Davis' tables, published by Longmans, Green & Co., will be found to be admirably suited for the work. Peabody's tables are preferable for some kinds of work, while Marks and Davis' tables will be found preferable for other kinds. All the values given in the book, except those specially noted, are from Marks and Davis' tables.

In the first edition of any work of this character, it is difficult to entirely eliminate errors, even by the exercise of the greatest care. Accordingly, the author will be very grateful to any of his readers who will point out to him errors of any kind which he may find, whether in the statements made or in the answers to the problems.

In conclusion the author wishes to express his thanks to many friends who have assisted in the preparation of this work, particularly Professors Charles James, L. S. Marks, and C. H. Peabody, Dr. William Kent and Mr. Geo. Orrok. He also wishes to acknowledge his indebtedness to the firms mentioned on page x for material furnished.

#14

ERRATA

Page 7. Fig. 1. Ordinates should be -0.20, -0.10, 0.0, $+0.10$, $+0.20$.

Page 11. Prob. 3. Ans. 3.2174. Prob. 13. Ans. 13.998.

Page 15. In equation (4) substitute R for $C'V$, not for C'.

Page 23. Prob. 4. Ans. 200.4.

Page 36. Last expression should be $\frac{\gamma}{\gamma-1} WP(V_2-V_1)$.

Page 37. Equation (2). Insert a minus sign before $\frac{dP}{P}$.

Page 38. Art. 51. P in equation should be P_1.

Page 46. Equation 6. Insert brackets after $\frac{\gamma}{\gamma-1}$ and at end.

Page 49. Prob. 16. Ans. 123,000. Prob. 18. Ans. 25 and 510. Prob. 27. Read 26 for 24.

Page 50. Prob. 40. Ans. 0.809. Prob. 41. Ans. 0.0081. Prob. 42. Ans. 0.0142. Prob. 43. Ans. 1.10.

Page 57. Line 3. Read 4 for 3.

Page 63. Prob. 6. Read ft.-lbs. for B.T.U. Prob. 15. Ans. 42.8.

Page 71. Prob. 3. Insert 161.1 in ans. Prob. 5. Ans. 333.0. Prob. 6. Ans. .002851. Prob. 9. Ans. 798.1.

Page 77. In paragraph 4, line 2, read 0.0886 and so in line 5.

Page 80. In equation read 1.4094.

Page 87. Prob. 6. Ans. 24.11. Prob. 7. Ans. 0.0415.

Page 88. Prob. 28. Ans. 10,495. Prob. 32. Ans. 1.6020.

Page 95. Prob. 7. Ans. 14.956.

Page 96. Prob. 16. After " air " insert "in Problem 15."

Page 108. Second line from bottom, read P' for P.

Page 130. In second equation read 25.36 for 25.0.

Page 139. Prob. 4. Ans. 25.4. Prob. 5. Ans. 26.6. Prob. 6. Ans. 1.064.

Page 140. Prob. 9. Ans. 99.7. Prob. 11. Ans. 12.9. Prob. 14. Ans. 122,800. Prob. 15. Ans. 15.7. Prob. 17. Cannot be solved since compression cannot be complete. Prob. 18. Cannot be solved.

Page 156. Prob. 1. Ans. 85. Prob. 5. Ans. 10.5. Prob. 10. Ans. 0.0276. Prob. 11. Read "cylinder" for "piston." Ans. 19.6.

Page 177. Prob. 2. Ans. 0.20 and 0.18. Prob. 3. Ans. 0.400 and 0.447. Prob. 9. Assume 120 r.p.m. Prob. 12. Ans. 23,100.

Page 185. In equations at bottom of page, read 1.1778 for 1.1178. Read 0.964 for 96.4. Read 1186.3 for 1183.3. Read 7.40 for 7.45. Read 0.726 for 0.505.

Page 198. Prob. 5. Read 17.9 for 16.9.

(*Over*)

Page 214. In article 223, paragraph 2, line 8, read seventh for sixth, ar in line 9 read sixth for seventh.

Page 215. In col. 11 read 11.60 for 11.50, in col. 12 read 8.93 for 8.83 and col. 14 read 2.748 for 2.848 and 3.48 for 1.212.

Page 217. Sixth line from bottom. For 0.217 read 0.189. Fourth line frc bottom. For 3.792 read 3.683.

Page 218. Fourth line. Read 3940 for 3830. Sixth line. Read 4030 for 3900.

Art. 226. Par. 2. Line 6. Equation should be $9 \times 1052.3 = 9471$. Line 10. Equation should be $62{,}032 - 9471 = 52{,}562$. Line 12. Read 3500 for 3290. Line 13. Read 3570 for 3360.

Page 219. Par. 3. Line 6. Equation should be $62{,}032 \times \frac{1}{13} = 4770$. Line 14. Read 18,170 for 18,250.

Page 232. Prob. 3. Ans. 21.1. Prob. 4. Ans. 2560. Prob. 6. Ans. 2120. Prob. 7. Line 2 read CO for CO_2.

Page 233. Prob. 19. Ans. 2970.

Page 251. Prob. 7. Ans. 60.3.

Page 256. Equation 6. Read .74 for .75.

Page 263. Prob. 4. Ans. 11.0.

Page 273. Prob. 7. Ans. 3.83. Prob. 8. Ans. 1290. Prob. 10. Ans. 15,100. Prob. 11. Ans. 3,130,000.

Page 290. Equation (2). For $\frac{140}{14.7}$ read $\frac{14.7}{140}$.

Page 295. Line 11. For P read E.

Page 301. Answers contain some errors in third significant figure.

Page 319. Prob. 2. Ans. 2.26. Prob. 3. Ans. 71,700. Prob. 4. Ans. 1002. Prob. 5. Ans. 573. Prob. 6. Ans. 40.5. Prob. 7. Ans. 232,500. Prob. 8. Ans. 160,800. Prob. 9. Ans. 13,660. Prob. 11. Ans. 2.52.

Page 343. Prob. 2. Ans. 41,400.

Page 344. Prob. 4. Ans. 6900. Prob. 5. Ans. 8590. Prob. 14. Ans. 0.0066. Prob. 15. Ans. 0.00503. Prob. 16. Ans. 503.

Page 367. Prob. 3. Ans. 32.5. Prob. 5. Ans. 45,150. Prob. 6. Ans. 181. Prob. 7. Ans. 251. Prob. 10. Ans. 113.

Page 369. Third line from bottom. Read isothermally for adiabatically.

Page 371. Second line from bottom. Read $b-c$ for $b-d$.

Page 383. Fifth line from bottom. Read *cdfg* for *idfg*.

Page 384. Last line of Art. 348. Read *cdfg* for *idfg*.

CONTENTS

ACKNOWLEDGMENTS

The author is indebted to the following firms for illustrations and other material in this book.

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PRACTICAL THERMODYNAMICS

CHAPTER I

INTRODUCTION—THE NATURE AND MEASUREMENT OF HEAT

1. The Purpose of Thermodynamic Machinery. One of the greatest, if not the greatest, of our engineering problems, is the transformation of the store of energy with which Nature is so lavishly endowed, into those forms which best serve the purpose of mankind. The form of energy which men find to be the most generally useful for their purposes is that form which we term work, or mechanical energy. Unfortunately, the forms in which energy is furnished to us by Nature are seldom those which are immediately available to our purpose, or which may be transformed into work by simple mechanical appliances, such as windmills or water-wheels. The needs of society are usually such that mankind is commonly obliged to avail himself of that vast store of natural energy found in the form of potential chemical energy in combustible substances. This form of energy can be liberated, so far as we know, only in the form of heat. In order to make Nature's store of energy of use to us, therefore, it is necessary first to transform it into the form of heat, and then in most cases to transform this heat into some more useful form of energy, such as work, or electricity, or light. In doing so, we make use of certain forms of engineering apparatus which we may term **thermodynamic machines.**

Thermodynamics, in the sense in which it is used by physicists, is that branch of physical science which treats of the effects produced by heat and the phenomena accompanying their various manifestations. When used by the engineer, however, the term thermodynamics is understood to mean that branch of engineering science which deals with the interconversion of heat and work, and the phenomena attendant thereon.

2. The Conservation and Correlation of Energy. One of the fundamental axioms of physical science is that the sum total of the energy

of the universe is a constant quantity and that this energy may not be increased or diminished, created or destroyed, by any known process or power. This physical axiom, known as the doctrine of the conservation of energy, lies at the basis of our theory of thermodynamics. As a corollary of the axiom of the conservation of energy, we may state that the different forms of energy are mutually inter-convertible, and that the amount of one form of energy which will be required to produce a given amount of any other form of energy is fixed and invariable. For instance we find that a certain quantity of work will invariably be transformed into a certain quantity of heat, that a certain quantity of electrical energy is the equivalent of a definite quantity of potential chemical energy, and so on throughout the entire list of possible conversions.

3. Standards of Measurement. Fundamental Units. As a preliminary to the intelligent discussion of any engineering subject, it is necessary to establish certain standards of measurement. Without such standards it is impossible to express quantitative relations, or to make of the physical sciences anything but an orderly array of curious and interesting, but generally useless, facts. If these standards of measurement are to be of value in engineering work, they must be those which society uses in its ordinary dealings, and with which mankind generally, and workmen more particularly, are thoroughly familiar. Accordingly, engineers in English-speaking countries use as their standards of measurement those units which are collectively known as the Foot Pound Second system. This system, while not so elaborate, or perhaps so rational, as the C.G.S. system in use among physicists, has the immense practical advantage that its units are understood by everyone, and are those in common use in our workshops.

4. Length. The unit of length is the foot which is defined [1] as 0.30480 meter or 30.48 centimeters. The meter is defined as the length of a certain bar of metal, accepted as the standard of length by international agreement. The centimeter, which is the standard of length used in physical measurements, is $^1/_{100}$ part of the length of this bar. In certain engineering work the inch is a unit of length. The inch is, however, never used in rational energy equations. The symbol of length is L.

5. Mass. The unit of mass, or quantity of matter, is the pound, which is defined [1] as 0.453592 kilogram, or 453.592 grams. A kilogram is the mass contained in a certain metal weight, accepted as the standard of mass by international agreement. The gram is $^1/_{1000}$ part of the kilogram. The symbol of mass, when expressed in pounds, is W.

[1] By Act of Congress.

6. Time. The unit of time is the mean solar second, invariably called simply the second (except in works on astronomy), which is $^1/_{86400}$ part of the mean solar day. The symbol of time is t.

7. Force. The unit of force is the weight of 1 pound, or the force with which the earth attracts a mass of 1 pound, at any point where the acceleration produced by gravitation is 32.1740 feet per second per second.[1] For convenience and brevity, the unit of force is also known as a pound. The symbol of force is F.

8. Relation between Force, Mass, and Acceleration. It will be noted that the unit of force (1 pound) does not produce an acceleration of 1 foot per second per second, in the unit of mass (1 pound). In consequence of this fact, in all kinetic energy equations we use for the unit of mass that quantity of matter to which the unit of force does give an acceleration of 1 foot per second per second. This quantity of matter is 32.1740 pounds and is usually termed the **kinetic mass unit.** The symbol for mass, when expressed in kinetic mass units, is M, which is, of course, equal to $\frac{W}{g_0}$.

9. Derived Units. From the four fundamental units of length, mass, time, and force, the following units of measurement are directly derived:

10. Area. The unit of area is the square foot. In practical engineering calculations the square inch, which is, of course, $^1/_{144}$ part of the square foot, is generally used as the unit of area. The symbol for area is A.

11. Volume. The unit of volume is the cubic foot. The symbol for volume is V.

12. Work, or Energy. The unit of work, and therefore the fundamental unit of energy, is the foot-pound, which is the quantity of work performed by a force of 1 pound in moving its point of application through a distance of 1 foot, measured in the direction of its line of action. The symbol for energy, when expressed in foot-pounds, is U.

13. Power. Power is the rate of doing work, or the rate of expenditure of energy. The practical unit of power is the horse-power, which is equivalent to the performance of work at the rate of 550 foot-pounds per second. The symbol for power, when measured in horse-power, is H.P.

14. Pressure. Intensity of pressure, called for brevity, pressure, is the rate of application of a uniformly distributed force upon an area,

[1] This value for the acceleration of gravity has been accepted as standard by international agreement. The symbol for the acceleration produced by gravity is g, and for the value 32.1740 the symbol g_0 may be used.

and is found by dividing the total force by the total area upon which it is applied. The units of pressure are four in number. The first is the pressure of 1 pound per square foot. This is the unit of pressure employed in all rational thermodynamic equations. The second is the pressure of 1 pound per square inch, which is 144 times as great as the first. This is the unit employed in engineering tables, such as steam tables, and so on, and in engineering calculations. The third is the pressure produced by a column of pure mercury 1 inch high, at a temperature of 32° F., when g is 32.1740 feet per second per second. This is the unit generally used in condenser and gas calculations. The fourth is the normal pressure of the atmosphere at sea level, which has been defined by international agreement to be the pressure produced by a column of pure mercury 29.921 inches high, at a temperature of 32° F. where g is 32.1740 feet per second per second. This unit of pressure is termed an atmosphere. The symbol for pressure, when expressed in pounds per square foot, is P.

15. Absolute and Gage Pressure. Pressure gages of the type ordinarily used in engineering work do not measure the true pressure exerted upon their mechanism, but the excess of this pressure over that of the atmosphere. The pressure recorded by such an instrument is called the gage pressure, and may be reduced to the true or absolute pressure by adding the actual pressure of the atmosphere as deduced from the barometer reading. In like manner, a vacuum gage registers the amount by which a pressure falls short of that of the atmosphere and the "vacuum" so recorded may be reduced to absolute pressure expressed in inches of mercury, by subtracting it from the reading of the barometer. Unless pressures are stated to be gage pressures, they are understood to be absolute pressures, in works of thermodynamics.

16. Effects of Heat. Energy itself cannot be perceived. We may only perceive and measure it by the effects which it produces. Heat, being a form of energy, can only be perceived and measured by its effects. We term a body hot or cold according to the effects which we feel when we are in contact with it or in its neighborhood. We find that hot bodies tend to give up heat to cold bodies, and eventually all attain the same degree of warmth, when they are brought near one another. When one body is capable of giving up heat to another body we say that the first body has a higher temperature than the second. As a result of heat exchange between bodies of different temperatures (i.e., the addition of heat to cold bodies and the abstraction of heat from hot bodies), we find that some or all of the following effects are produced.

First, the addition of heat to a body almost always increases its

temperature, making it more capable of giving up heat to colder bodies, and less capable of absorbing heat from hotter bodies. The rise in temperature is usually very nearly proportional to the quantity of heat added. Second, the addition of heat to a body, with consequent rise in temperature, generally tends to expand that body, and the amount of the expansion is usually very nearly proportional to the quantity of heat added. Third, the addition of heat to a body, with consequent rise in temperature, quite often results in a change in the physical state of the body. For instance, the addition of heat to ice changes it into water; the addition of heat to water changes it to steam.

The abstraction of heat has in each of the above cases the contrary effect, reducing the temperature, decreasing the volume, and changing the physical state from that of a gas or liquid to that of a liquid or solid. For instance, the abstraction of heat from carbon dioxide, which is a gas, reduces its temperature and volume, and finally changes it to a snow-like solid.

The addition of heat to certain kinds of bodies results in a change in their chemical composition. For instance, the addition of heat to potassium chlorate, $KClO_3$, changes it to potassium chloride and oxygen, the formula for the reaction being $2KClO_3 = 2KCl + 3O_2$. Such phenomena, however, unlike the ones recorded in the preceding paragraph, are not usually reversible. Subsequent abstraction of heat will not return such substances to their original chemical form.

In all of these cases, the amount of change produced, as measured by the quantity of material changed, is always proportional to the amount of heat producing the change, so that heat may be, and is, measured quantitatively by scientists, by means of the physical effects which it produces.

17. The Measurement of Temperature. Since the most obvious change ordinarily produced in a body by the addition or abstraction of heat is the elevation or depression of its temperature, we must first seek some suitable method of measuring temperature. It has been found that ice melts at a certain definite temperature, provided that the ice be formed from pure water, and the fusion occurs at a pressure of one atmosphere. This temperature is known in physics as the ice-point. It is also known that the temperature of the steam which comes from boiling water at a pressure of one atmosphere is a fixed quantity. This temperature is known in physics as the boiling-point. If we may find some method of determining the temperature of a body with reference to these two points, we will have a system of thermometry, or temperature measurement.

We have noted that one of the effects of heat is to expand almost

all bodies to which it is added. Gases expand to a greater degree upon the addition of heat than do any other substances, and are therefore better suited than other substances to the purposes of exact thermometry. Some gases expand with great regularity, the amount of expansion so produced being strictly proportional to the quantity of heat producing the expansion and giving a definite measure of the rise in temperature of the gas. Helium and hydrogen are such gases. Other gases, such as air, oxygen, etc., while often used in thermometry, are less regular in their rate of expansion. Still other gases, such as carbon dioxide, are so irregular in their rate of expansion as to be quite unsuited for the purposes of thermometry. It is therefore the custom among physicists, as a result of international agreement, to adopt for the measurement of temperature the indications of a hydrogen thermometer.[1] The method of construction and use of such a standard thermometer is described in Art. 31. The symbol of temperature is T. (See Art. 26.)

18. Thermometer Scales. In engineering work in English-speaking countries the Fahrenheit thermometer scale is in common use. One degree on the Fahrenheit scale is defined as $1/180$ part of the rise in temperature, as indicated by a standard hydrogen thermometer, from the ice-point to the boiling-point. The Fahrenheit zero is 32° below the melting-point of ice, so that the temperature of the ice-point is 32° Fahrenheit (generally written 32° F.) and the temperature of the boiling-point is 212° F. In countries using the metric system, and in purely scientific work, the centigrade thermometer scale is used. The centigrade degree is $1/100$ part of the rise in temperature from the ice-point to the boiling-point. The centigrade zero is the ice-point and the centigrade temperature of the boiling-point is 100° (generally written 100° C.).

In order to reduce Centigrade to Fahrenheit temperatures, we may make use of the formula

$$F = \frac{9C}{5} + 32. \qquad (1)$$

In order to reduce Fahrenheit to Centigrade temperatures, we may make use of the formula

$$C = \frac{5(F - 32)}{9}. \qquad (2)$$

[1] For reasons of convenience, it is not the volume of the hydrogen, but the pressure which it exerts upon the walls of the bulb in which it is confined at constant volume, which affords a measure of its temperature.

In these formulæ, C and F are respectively corresponding Centigrade and Fahrenheit temperatures.

19. Mercury Thermometers. For the ordinary measurement of temperature in engineering work, "mercury in glass" thermometers are used. The indications of a perfect thermometer of this type, (i.e., one filled with pure mercury, and having a capillary tube of absolutely uniform bore) depend upon the kind of glass from which it is made, and the conditions under which it is used, and are invariably different from those of the hydrogen thermometer, except at the ice-point and boiling-point. Such thermometers are, however, sufficiently exact for most engineering work, although entirely unsatisfactory for refined investigations, unless suitably handled and calibrated. In Fig. 1 will be found a graphical representation of the errors of a perfect mercury in glass thermometer at different temperatures. It will be seen that the error is so small as not to be important in ordinary engineering work.

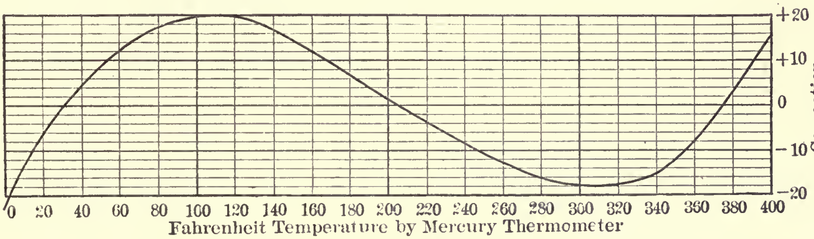


FIG. 1.—Correction curve to reduce mercury in glass thermometer readings to hydrogen thermometer readings.

20. The British Thermal Unit. Temperature measures intensity of heat, or in other words, it determines the ability of a body to surrender heat to, or abstract it from, a body of a different temperature. It does not, however, tell us the amount of heat which the body contains. In order to measure quantity of heat, we need another unit of measurement aside from that of temperature. The unit of measurement used in engineering calculations in English-speaking countries is termed the Mean British Thermal Unit, and may be defined as $^1/_{180}$ part of the heat required to raise 1 pound of pure water from a temperature of 32° F. to a temperature of 212° F. (i.e., from the ice-point to the boiling-point) without loss of mass, under a pressure of one atmosphere. In order to avoid the use of the cumbersome term Mean British Thermal Unit, the symbol B.T.U. is used. The symbol for quantity of heat is H.

21. The Mechanical Equivalent of Heat. Since heat is a form of energy and the different forms of energy are inter-convertible, it follows that work may be converted into heat. As a result of a long series of experiments carried out by many different men at various times, it has been shown that 777.5± foot-pounds of work may be transformed into one B.T.U. This quantity, 777.5 foot-pounds, is

known as the mechanical equivalent of heat, and in thermodynamic equations, we use for its exact (but unknown) value, the symbol J.

22. Specific Heat. If a small quantity of heat be added to a substance without changing its physical or chemical state, its temperature will be somewhat increased. The rise in temperature produced will be directly proportional to the quantity of heat absorbed, and inversely proportional to the mass absorbing it. The proportionality factor, which varies widely for different substances, is known as the **specific heat** of the substance. The specific heat of a substance may be defined as the number of B.T.U. required to raise the temperature of 1 pound of the substance 1° F. It follows from the definition of the Mean British Thermal Unit, and that of specific heat, that the average specific heat of water between the temperature of 32° and 212° F. is unity. The term average is used, because it has been found that the specific heat of different substances, water included, is not a constant quantity, but depends upon the temperature of the substance. Therefore, in physical or engineering investigations requiring great exactitude, account must be taken of this variation in the specific heat of substances, although in ordinary engineering work it is not necessary to do so.

Tables are appended showing the relation of the English, the Metric, the Electrical Engineering, and the C.G.S. systems of units, used in engineering and physical measurements.

Table I

LENGTH

	Feet.	Inches.	Meters.	Centimeters.
Foot	1	12	0.30480	30.48000
Inch	0.08333	1	0.02540	2.54001
Meter	3.28083	39.37996	1	100
Centimeter	0.032808	0.39380	0.01	1
Millimeter	0.003281	0.03938	0.001	0.1

Table II

AREA

Sq.	Feet.	Inches.	Meters.	Centimeters.
Foot	1	144	0.092903	929.03
Inch	0.0006944	1	0.00064516	6.4516
Meter	10.7639	1550.0	1	10,000
Centimeter	0.00107639	0.15500	0.0001	1

TABLE III

VOLUME

Cubic.	Feet.	Inches.	Yards.	Meters.	Liters.	Centimeters.
Foot......	1	1728	.037038	.028317	28.317	28317
Inch......	.0005787	1	.000021433	.000016387	.016387	16.387
Yard......	27	46,656	1	.76454	764.54	764.540
Meter.....	35.314	61,023	1.3081	1	1000	1,000,000
Liter......	.035314	61.023	.001308	.001	1	1000
Centimeter	.000035314	.061023	.000001308	.000001	.001	1

TABLE IV

MASS

	Pounds.	Kilograms.	Grams.
Pound.....................	1	0.453592	453.592
Kilogram..................	2.04622	1	1000
Gram......................	0.00204622	0.001	1

TABLE V

FORCE

	Pounds.	Kilogram.	Dyne.
Pound.....................	1	0.453592	444,800
Kilogram..................	2.04622	1	980,665
Dyne......................	0.0000020866	0.00000101975	1

TABLE VI

PRESSURE

	Pounds per sq.in.	Pounds per sq.ft.	Inches of Hg.	Atmospheres.	Kilos per sq.cm.	Millimeters of Hg.	Dynes per sq. cm.
1 pound per sq.in.	1	144	2.0360	0.068044	0.070307	51.713	68,947
1 pound per sq.ft.	0.0069441	1	0.014140	0.00047255	0.00048825	0.35920	478.80
1 inch of Hg	0.49117	70.721	1	0.033420	0.034532	25.400	33,863
1 atmosphere	14.696	2116.3	29.921	1	1.0333	760	1.013300
1 kilo per sq.cm	14.223	2048.2	28.958	0.9678	1	735.54	980,665
1 mm. of Hg	0.019337	2.7840	0.03937	0.0013158	0.0013595	1	1333.3
1 dyne per sq.cm	0.000014505	0.002106	0.000026532	0.0000009868	0.0000010198	0.00075000	1

TABLE VII

ENERGY

	Foot-pounds.	B.T.U.	Kilogram-meters.	Greater Calories.*	Joules.†	Kilowatt hours.	Horse-power Hours.
Foot-pound	1	0.0012861	0.13826	0.0003241	1.3558	0.00000037662	0.00000050505
B.T.U.	777.5	1	107.50	0.25200	1054.2	0.0002928	0.0003927
Kilogram-meter	7.2330	0.009302	1	0.0023442	9.80665	0.0000027244	0.0000036532
Greater calorie	3086.0	3.9683	426.6	1	4183.4	0.0011621	0.0015583
Joule	0.73756	0.0009486	0.10197	0.0002390	1	0.00000027778	0.00000037251
Kilowatt hour	2,655,200	3415	367,080	860.5	3,600,000	1	1.3410
Horse-power hour	1,980,000	2547	273,740	641.7	2,684,500	0.7457	1

* 1 greater calorie = 1000 lesser calories = .01 kilogram of water from 0° to 100 °C.

† 1 Joule = 10,000,000 Ergs.

PROBLEMS

1. With what force does the earth attract a mass of 1 lb., at a point where the acceleration produced by gravity is 32.1500 ft. per sec.? Ans. $\frac{32.1500}{32.1740}$ lbs.

2. Find the acceleration of gravitation at a point where a mass of 1 ton is attracted by the earth with a force of 2001.5 lbs. Ans. 32.1981 ft. per sec.^{2}

3. What acceleration will a force of 1 lb. impart to a mass of 10 lbs.?

Ans. 3.1274 ft. per sec.^{2}

4. What acceleration will a force of 1 lb. impart to a body whose mass is 2 kinetic mass units? Ans. 0.5 ft. per sec.^{2}

5. What will be the weight of a kinetic mass unit at a point where the acceleration of gravitation is 32.1600 feet per sec.^{2}? Ans. 32.16 lbs.

6. A piston is 10 ins. in diameter. Find its area. Ans. 0.5454 sq.ft.

7. A box is 18 ins. wide, 24 ins. long, and 12 ins. deep. What is its volume?

Ans. 3 cu.ft.

8. What work is performed in raising a mass of 100 lbs. a vertical distance of 10 feet at a point where $g=32.100$. Ans. $\frac{32{,}100}{32.174}=997.7$ ft.-lbs.

9. What power is required to raise a mass of one ton with a velocity of $5\frac{1}{2}$ feet per second, where $g=32.1740$? Ans. 20 H.P.

10. The weight of a cubic inch of ice-cold mercury is 0.491170 lb. What pressure in pounds per square inch equals 10 ins. of mercury (correction for variation in the force of gravity omitted)? Ans. 4.9117 lbs. per sq.in.

11. What is the value of an atmosphere in pounds per square inch?

Ans. 14.696 lbs. per sq.in.

12. What in pounds per square foot? Ans. 2116.3 lbs. per sq.ft.

13. A pressure gage reads 60 lbs. when the pressure of the air is 14.5 lbs. per square inch. What is the absolute pressure? Ans. 74.5 lbs. per sq.in.

14. The barometer reads 28.5 ins. What is the absolute pressure of the air?

Ans. 14.007 lbs. per sq.in.

15. What is the absolute pressure in a condenser when a vacuum gage attached to it reads 26 ins. with the barometer as in Problem 14?

Ans. 2.5 ins. = 1.229 lbs. per sq.in.

16. The temperature of a body is 50° F. Find its Centigrade temperature.

Ans. 10° C.

17. The temperature of a body is 20° C. Find its Fahrenheit temperature.

Ans. 68° F.

18. A mercury thermometer indicates a temperature of 60°. What is the temperature by the hydrogen thermometer? Ans. 60.12° F.

19. What quantity of heat will be required to raise 1 ton of water from the ice-point to the boiling-point. Ans. 360,000 B.T.U.

20. What is the mechanical equivalent of 10 B.T.U.? Ans. 7775 ft.-lbs.

21. 1000 ft.-lbs. of work are the equivalent of what quantity of heat?

Ans. 1.2850 B.T.U.

22. A body weighing 3.5 lbs. is raised 4° in temperature by the expenditure of 7 B.T.U. Find the specific heat. Ans. 0.5.

23. A body weighing 20 lbs. and having a specific heat of 0.700 is raised 10° in temperature. What quantity of heat is absorbed? Ans. 140 B.T.U.

CHAPTER II

THE THERMAL PROPERTIES OF GASES

23. The States of Matter. All substances may be classified according to their physical state as solid or fluid. Solid substances are those which are not permanently deformed by slight forces and which therefore retain their form when they are unconfined. Fluids are those substances which flow, that is, which have no definite form except that which is imposed upon them by the containing vessel. Fluids are classified as elastic and inelastic. Inelastic fluids or liquids, are those which are practically incompressible, and therefore do not change appreciably in volume or density under varying pressures. Elastic fluids are in turn divided into vapors and gases. Vapors are those elastic fluids which are readily transformed into liquids by a slight reduction in temperature. Gases are those elastic fluids which require a considerable reduction in temperature to reduce them to the form of liquids. As an example of these various physical states, we may cite iron as a solid, water as a liquid, steam as a vapor, and air as a gas.

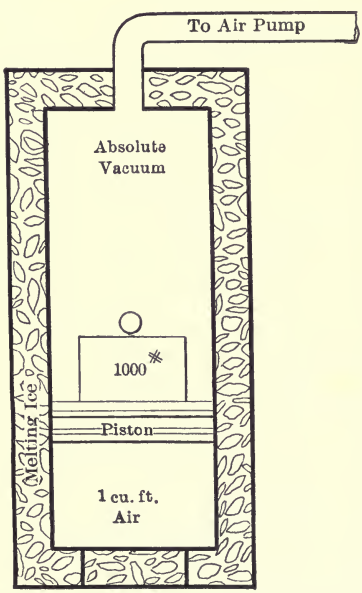


FIG. 2.—Ideal apparatus for the investigation of the properties of gases.

24. The Relation between the Pressure and Volume of a Mass of Gas. Let us assume that we have a receiver whose construction is such that we may vary its volume at will, and that it contains a quantity of air. Such a receiver is shown in Fig. 2. This imaginary apparatus consists of a cylinder about $13\frac{1}{2}$ inches in internal diameter and having a cross-sectional area of 1 square foot. It is flat bottomed, and sliding within it there is a gas-tight piston without weight, which works smoothly up and down, without friction. Weights may be placed on the piston and the amount of weight in pounds so

placed, will obviously be the pressure of the confined gas, expressed in pounds per square foot. The distance between the bottom and the flat face of the piston, in feet, is equal to the volume of the confined gas, in cubic feet. The pressure of the air is removed from the top of the piston by a suitable air-pump. Let us assume that this receiver is surrounded by melting ice so that its temperature and that of its contents is kept at 32° F. Let us assume also that the quantity of air which the receiver contains is such that when its volume is exactly 1 cubic foot (i.e., when the piston is 1 foot from the bottom) its pressure, measured above an absolute vacuum by the weights placed upon the piston, will be exactly 1000 pounds per square foot. If now we place 2000 pounds upon the piston, it will be forced downward until the volume of the confined air becomes $\frac{1}{2}$ a cubic foot. If the weight upon the piston be made 500 pounds, the piston will rise until the volume of the confined air becomes 2 cubic feet, and if the weight be made $333\frac{1}{3}$ pounds, the volume will increase to 3 cubic feet.

An inspection of the foregoing figures will serve to show that the product of the pressure and volume is always 1000 foot-pounds. No matter how the pressure may be altered, the volume will so adjust itself that it will be inversely proportional to the pressure, and as long as no air enters, or escapes from, the receiver, and the temperature remains at 32° F., the product of the pressure and the volume will remain constant. We may express this mathematically by the equation

$$P\,V = P_0 V_0,$$

where P is the pressure of a confined mass of air at a temperature of 32° F.;
V is the corresponding volume of this mass of air;
P_0 is the original pressure of the same mass of air, at the same temperature;
V_0 is the original volume.

The above statement is subject to modification at very high pressures. All gases, air included, when sufficiently compressed and cooled, are changed first into vapors and then into liquids, in which states they behave in a very different manner. Therefore, when a gas is compressed to such a degree that it becomes a vapor, or its volume begins to approach the volume of the liquid into which it would condense, it no longer conforms to the above law, but its behavior approximates that of a vapor or a liquid.

Had we taken another gas instead of air—hydrogen, for instance—we would have found exactly the same relation between the volume and the pressure, provided we started with one cubic foot of gas at a pressure of 1000 pounds per square foot. No matter what the gas with

which we experiment, so long as the temperature be kept constant at 32° F., if the product of the pressure and volume be initially 1000 foot-pounds, this product will remain 1000 foot-pounds, no matter what we may subsequently make the volume or the pressure. The mathematical expression for the relation of the pressure and volume of any gas at a constant temperature is therefore the same as that already given for air, namely,

$$P\,V = P_0 V_0.$$

This law is known in physics as Boyle's law.

25. The Relation between the Pressure, Volume, and Temperature of a Mass of Gas. Let us now see what effect will be produced upon the pressure of a given mass of gas, confined in a given volume, by a change in its temperature. Consider again the ideal apparatus already described, containing 1 cubic foot of air at a pressure of 1000 pounds per square foot and a temperature of 32° F. If we raise the temperature of this mass of air while we keep the volume constant, its pressure will be found to increase. When the temperature has been raised by 491.6° (i.e., until it becomes 523.6° F.) the pressure will be increased by 1000 pounds, becoming 2000 pounds per square foot. Upon raising the temperature another 491.6°, to 1015.2° F., the pressure will increase another thousand pounds, rising to 3000 pounds per square foot. If by some means we depress the temperature to 213.8° below zero, the pressure will fall to 500 pounds per square foot.

26. The Absolute Zero. It will be apparent from the above facts that the rate of pressure increase or decrease, under the above circumstances, is 1000 pounds per square foot for each 491.6° increase or decrease of temperature. Since the pressure is by hypothesis, 1000 pounds per square foot at 32° F., at a temperature 491.6° lower, or −459.6°, the pressure will become zero. Had the original pressure at 32° F. been P pounds per square foot, the increase or decrease would have been found to be P pounds per square foot for every 491.6° increase or decrease of temperature, and the pressure would have become zero at −459.6°, no matter what the value of P might have been at 32° F.

The fact that the product of the pressure and volume of a gas tends, in theory at least, to become zero at −459.6°, led physicists to infer that a body having this temperature would be devoid of heat energy, and therefore could not possibly have a lower temperature. Many other phenomena have since been noted, which point to the same conclusion, so that this temperature has become known in physics and thermodynamics as absolute zero. The absolute temperature of a body is then its temperature measured above absolute zero, and is found by adding 459.6° to the Fahrenheit temperature. (In case the absolute tem-

perature is wanted in Centigrade degrees, add 273.1° to the Centigrade temperature.)

The lowest temperature which has been actually obtained is −457° F., which is about 3° absolute. At this temperature all known gases become liquid at ordinary pressures, and most of them solidify. Metals become intensely brittle, and also almost perfect conductors of electricity. Many strange magnetic, electrical, and chemical phenomena are noted. This temperature was obtained by the condensation of helium at high pressure, and its subsequent evaporation under an air-pump.

27. The Characteristic Equation of Gases. An inspection of the figures given in Art. 26 shows that the pressure of a mass of gas confined within a constant volume is proportional to the absolute temperature of the gas. We may express this fact mathematically by the equation,

$$P = C'T, \qquad \qquad (1)$$

where P is the pressure of the gas,

T is the absolute temperature, and

C^1 is a constant depending on the composition, mass, and volume of the gas.

Let us assume that the mass of the gas is 1 pound. Multiplying both sides of equation (1) by V we have

$$P\,V = C'V\,T, \qquad \qquad (2)$$

where V is the volume in which the pound of gas is confined. Now since by hypothesis the gas is maintained at constant volume, we may write

$$C'V = R, \qquad \qquad (3)$$

where R is a constant independent of T, since both C' and V are constants.

Substituting R for C' we have

$$P\,V = R\,T, \qquad \qquad (4)$$

for 1 pound of any gas at a constant volume. Now it has already been shown in Art. 24 that when T is a constant the product PV is also a constant. Therefore the value of R for any gas is a constant, and is independent of the value of V or P as well as of T.

Since the mass of a quantity of gas at a given pressure and temperature is proportional to its volume, it follows that the equation relating the pressure, temperature, volume, and mass of a quantity of gas may be written

$$P\,V = W\,R\,T, \qquad \qquad (5)$$

where P = the absolute pressure of the gas in pounds per sq.ft.;
V = the volume of the gas in cubic feet;
W = the mass (or weight) in pounds of the quantity of gas considered;
T = the absolute temperature of the gas, on the Fahrenheit scale, found by adding 459.6° to the Fahrenheit temperature of the gas; and
R = a constant depending upon the density of the gas. (The density depends on the chemical constitution of the gas, i.e., on the quantity and molecular weight of its constituents.)

28. Perfect Gases. A gas whose properties are such that it fulfills exactly the relation

$$P V = W R T,$$

at all temperatures and pressures, is called a **perfect gas.** It is now known that no gases are perfect in this sense. However, at ordinary temperatures (i.e., from −100° to +600° F.) and at pressures of less than ten atmospheres, the irregularities in behavior of such gases as hydrogen, helium, etc., are so inconsiderable as to be detected only by the most refined physical measurements, and such gases are therefore said to be sensibly perfect under such conditions. Practically all gases are so nearly perfect that they may be considered perfect gases in engineering computations.

A quantity of gas having a definite pressure, temperature, volume, and mass is said to have a definite thermodynamic state which is defined by these quantities. It will be noted that there are four quantities upon which the state of a gas depends, and if three of them are known, the fourth may be found. The expression

$$P V = W R T$$

defines the relation of these quantities which determine the state of the gas.

29. A Second Form of the Characteristic Equation. If we assume that a certain mass of gas has a pressure P_0, a volume of V_0, and a temperature T_0, we may write

$$P_0 V_0 = W R T_0. \qquad (1)$$

Transposing, this may be written

$$W R = \frac{P_0 V_0}{T_0}. \qquad (2)$$

If this quantity of gas suffers a change of state, and, as a result, assumes a pressure P, a volume V, and a temperature T, we may of course write

$$P V = W R T, \quad \ldots \ldots \ldots \ldots \quad (3)$$

and therefore by transposition,

$$\frac{P V}{T} = W R. \quad \ldots \ldots \ldots \ldots \quad (4)$$

Combining equations (2) and (4) we have

$$\frac{P V}{T} = \frac{P_0 V_0}{T_0}. \quad \ldots \ldots \ldots \ldots \quad (5)$$

This equation is often more convenient to use than the equation developed in Art. 27. So long as P and P_0, V and V_0, and T and T_0 are in the same units, they may be units of any convenient size. For instance, we may, in using this expression, measure pressures in millimeters of mercury, volumes in liters, and temperatures in Centigrade degrees on the absolute scale. So long as the same units are employed throughout the equation, it makes no difference what these units may be.

30. Value of R in the Characteristic Equation. Since at given pressures and temperatures, different gases have different densities (i.e., weights per cubic foot), it follows that their volumes per pound and therefore the value of R found from the expression

$$R = \frac{P V}{W T},$$

will differ, the value of R being inversely proportional to the density, and therefore to the molecular weight of the gas.

The best value for the density of the air free from water vapor and carbon dioxide, as given by Traverse, from Moreley, Rayleigh, and Leduc, is 1.29284 grams per liter, at 32° F. under a pressure of one atmosphere. Reducing to the F.P.S. system we find this to be 0.080710 pounds per cubic foot. Since one atmosphere equals 2116.3 pounds per square foot, we may write in the expression

$$P V = W R T,$$

$$2116.3 \times 1 = 0.080710 \times R \times 491.60$$

and

$$R = 53.338 \text{ for air.}$$

The density of oxygen is, from the same authority, 1.42900 grams per liter. Since the function R varies inversely as the density of the gas, we will find R for oxygen to be

$$R = 53.338 \times \frac{1.29284}{1.42900} = 48.256.$$

The molecular weight of oxygen gas is 32. The value of the function R for oxygen, as we have just seen, is 48.256. Since the value of R is inversely proportional to the molecular weight of any gas, which we may designate by the symbol M, we may write

$$\frac{48.256}{R} = \frac{M}{32},$$

from whence

$$R = \frac{48.256 \times 32}{M},$$

or

$$R = \frac{1544.2}{M}.$$

From the above equations it may be seen that the value of R for any gas of known chemical composition may be found by dividing 1544.2 by the molecular weight of the gas. Thus the molecular weight of CO_2 is 44. Hence the value of R for CO_2 is $1544 \div 44 = 35.11$. If the molecular weight of the gas is unknown, but the density, as compared with that of air, is known, the value of R may be found by dividing 53.338 by the density of the gas. Thus the density of CO_2 is 1.520, whence the value of R is $53.338 \div 1.520 = 35.11$.

For the density, value of R, and other thermodynamic properties of the principal gases, see Table VIII on page 23.

31. Gas Thermometers. The gas thermometer depends upon the principle that the product of the pressure and volume of a sensibly perfect gas is proportional to its absolute temperature. Air, nitrogen, helium, and hydrogen are the gases usually employed in such instruments. The thermometers are of two types, first those in which the gas is confined within a bulb having a definite volume, and the temperature is measured by measuring the pressure exerted by the gas, and second, those in which the gas is confined under constant pressure and the temperature is measured by finding the amount of expansion or contraction which the gas undergoes. It has been found that hydrogen and helium are the two gases which give the best results when used as thermometric fluids, and there seems to be no difference in behavior which would indicate that one was better than the other. Accordingly, physicists have adopted for the standard of temperature measurement a constant volume hydrogen thermometer, in which the gas is confined at a pressure of 1000 millimeters of mercury when its temperature is zero Centigrade.

Such a thermometer is shown in principle in Fig. 3. In this drawing, A is a bulb of glass, fused quartz, or other suitable material, in which the gas is confined. To this bulb is sealed a capillary tube B, in which the mercury may be made to rise by means of the reservoir R connected to B by the rubber tube C. The tube D, from the top of which the air has been exhausted, is also connected to B, and acts as a barometric tube. The difference between the level of the mercury in the reservoir and

in the tube D will, of course, be the barometric pressure of the atmosphere. The difference in level between the mercury in the capillary tube and in the tube D will be the absolute pressure of the gas confined in bulb A, which will be sensibly proportional to the absolute temperature of this gas.

In using the instrument, the bulb A is immersed in a bath whose temperature is to be measured. The reservoir R is elevated until the mercury is brought to a definite point, which is marked on the capillary tube. The difference in level, l, is then read by means of a cathetometer or some similar instrument. Allowances must, of course, be made for the thermal expansion of the material of the bulb, for the fact that the gas in the capillary tube B is not of the same temperature as the gas in the bulb, for the expansion, or contraction, of the bulb produced by the difference in pressure of the bath in which it is immersed and the gas which it contains, for the density of the mercury column, which is affected by its temperature, and for the value of the gravitational constant at the place where the measurement is made.

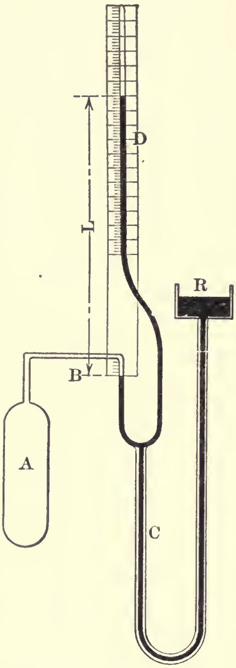


Fig. 3.—Diagrammatic sketch of a constant volume thermometer.

32. The Specific Heats of Gases. When a gas is heated the amount of heat required is found to depend, as in all other substances, upon the rise in temperature and upon the quantity of gas heated. Since an unconfined mass of gas does not occupy a definite volume, we may have two conditions under which a mass of gas may be heated. It may be confined within a definite space and heated at constant volume, or it may be confined under a definite pressure and allowed to expand as it is heated. In case the gas is confined at constant volume, the increase of pressure resulting from the addition of heat does not, of course, perform work. When, however, the gas is confined at constant pressure, and allowed to expand as it is heated, work is performed. It is found that the amount of heat required to raise 1 pound of a gas 1° in temperature differs under these two circumstances, being greater when the gas is heated at constant pressure than when it is heated at constant volume. Theory indicates, and experiment shows, that the excess of heat required in the former case is equal to the amount of work done by the gas in expanding at constant pressure.

The specific heat of a substance has already been defined as the number of B.T.U. required to raise 1 pound of the substance 1° F. in temperature. In this definition nothing was said about pressure, but the pressure was tacitly assumed to be a constant pressure of one atmosphere. Since all substances expand somewhat on heating, it follows that a certain amount of the heat applied to any substance,

under such conditions, is transformed into work, when the substance expands against atmospheric pressure. Since, however, the amount of expansion of most substances is very small, the amount of work so done is also exceedingly small. In the case of iron, the amount of work done amounts to about 0.00009 per cent of the heat required, or the result is affected by nine parts in ten million. In the case of a gas, however, the increase in volume produced by heating is very considerable as compared with that produced in other substances, and therefore we are obliged to consider a gas as having two specific heats, one at constant pressure, and the other at constant volume.

The specific heat of a gas at constant pressure is, of course, the number of B.T.U. required to raise the temperature of 1 pound of it 1° F. at any constant pressure. This quantity is found to be independent of the amount of the pressure, so long as the pressure remains constant. The symbol for the specific heat of the gas at constant pressure is C_p. The specifie heat of a gas at constant volume is the number of B.T.U. required to raise the temperature of 1 pound of the gas 1° F. at constant volume. The symbol for the specific heat at constant volume is C_v. The specific heat at constant pressure is equal to the specific heat at constant volume plus the heat equivalent of the work done by 1 pound of the gas in expanding at constant pressure while rising 1° in temperature.

We may also express the specific heat of a gas in dynamic units (i.e., in foot-pounds). The specific heat of a substance in dynamic units is the number of foot-pounds required to raise its temperature 1° F., and is, of course, equal to 777.5 times its specific heat in B.T.U. per pound. The symbol for the dynamic specific heat of a gas at constant volume is K_v, and at constant pressure K_p. We may write then the two equations

$$K_v = J\,C_v, \qquad (1)$$

and

$$K_p = J\,C_p, \qquad (2)$$

33. The Relation of the Specific Heats and the Density. Assume a mass of gas weighing 1 pound to be confined at a temperature T within a volume V_1 at a pressure P. The volume, as may be readily seen, is

$$V_1 = \frac{R\,T}{P}, \qquad (1)$$

If this mass of gas be raised 1° in temperature at constant pressure, its final volume will become

$$V_2 = \frac{R\,(T+1)}{P}, \qquad (2)$$

The change in volume will be

$$V_2 - V_1 = \frac{R\,(T+1)}{P} - \frac{R\,T}{P}, \quad \ldots \ldots \quad (3)$$

and therefore, after multiplying by P we have,

$$P(V_2 - V_1) = R(T+1-T) = R, \quad \ldots \ldots \quad (4)$$

Now the first term of the equation is the work done by the pound of gas in expanding at constant pressure while the gas is rising 1° in temperature, which, as we have just seen, is equal to the difference in the two dynamic specific heats of the gas. Hence we may write

$$R = K_p - K_v = J\,(C_p - C_v), \quad \ldots \ldots \quad (5)$$

34. The Ratio of the Specific Heats. The ratio of the specific heat of a gas at constant pressure to its specific heat at constant volume is a function very much used in thermodynamic work, and is designated by the Greek letter γ. Its value may be expressed mathematically by the equation

$$\gamma = \frac{C_p}{C_v} = \frac{K_p}{K_v}, \quad \ldots \ldots \quad (1)$$

The values of γ and of C_v (and therefore of K_v) are difficult of direct determination. The values of R and C_p (and therefore of K_p) are easy of direct determination. Therefore, the values of γ and C_v are best determined by computation from the values of C_p and R, quantities which may be obtained with great accuracy from density determination and calorimetry measurements. It has already been shown that

$$R = K_p - K_v, \quad \ldots \ldots \quad (2)$$

Hence

$$K_v = K_p - R, \quad \ldots \ldots \quad (3)$$

and

$$\gamma = \frac{K_p}{K_p - R}, \quad \ldots \ldots \quad (4)$$

The values of γ and C_v given in the table on page 23 were computed from the above equations.

It will be apparent from the relations just shown that the two specific heats and therefore the function γ of a perfect gas are constant quantities. In the case of an imperfect gas, however, it will be seen that since R is not the same under all conditions, the values of one or both of the

specific heats, and of the function γ, will vary as the conditions of temperature and pressure are changed.

35. The Intrinsic Energy of a Gas. When a gas is heated at constant volume the amount of heat transferred to it is utilized entirely in raising the temperature of the gas, and therefore in increasing the quantity of energy which the gas possesses. The energy so added is called intrinsic energy, since it resides within the gas, and is not transferred by the gas to any other body.

When a gas is heated at constant pressure, we have seen that some of the energy imparted to it is expended in doing external work, i.e., in pushing back the confining walls. The intrinsic energy contained in the gas after a given rise in temperature is the same whether the gas be heated at constant volume or constant pressure, but the heat required in the latter case is greater than the intrinsic energy imparted to the gas by the amount of external work done. It will be seen then that the intrinsic energy of 1 pound of a perfect gas [1] is equal to the dynamic specific heat of the gas at constant volume, multiplied by the absolute temperature of the gas. Also if the external work performed by 1 pound of such a gas, while expanding at constant pressure from a temperature of absolute zero, be added to its intrinsic energy, their sum will equal the product of the dynamic specific heat at constant pressure into the absolute temperature of the gas.

36. The Joule-Thompson Effect. It follows from the above discussion of intrinsic energy that when a perfect gas expands without performing external work that its temperature will remain unchanged, since its intrinsic energy remains unchanged. This statement is known as Joule's law. The truth of this law may be tested by allowing a gas to flow from a region of high pressure to a region of low pressure through a porous plug of very considerable thickness, as for instance, a small tube tightly packed with cotton for a length of 6 inches or more. It has been found that under such circumstances the difference in temperature of the gas before and after expanding is very small. In the case of some gases, such as hydrogen, nitrogen, and the monatomic gases, it amounts, at the most, to a few hundredths of a degree per atmosphere difference in pressure. Were the gases perfect, there would be absolutely no change in temperature. The amount of deviation from Joule's law marks the degree of imperfection of the gas. A careful study of the properties of gases, conducted by many men and extended over many years, shows that so far as engineering calculations are concerned, that oxygen, hydrogen, nitrogen, and the monatomic gases and mixtures of these gases are sensibly perfect at ordinary temperatures.

In further proof of this fact we may note that the differences in the temperatures indicated by thermometers using these different gases under identical conditions are at the most only about $^1/_{100}$ part of a degree Fahrenheit, between the ice-point and the boiling-point, and are less than $^1/_{10}$ of a degree (i.e., about $^1/_{100}$ of one per cent) at the extreme temperatures for which such instruments are usually used. However, gases which are sensibly perfect at ordinary temperatures are not neces-

[1] I.e., one that is perfect at all temperatures.

rily even approximately perfect at very low or very high temperatures, although ley are assumed to be so, for purposes of convenience, in many engineering alculations.

The characteristic equations for imperfect gases and an account of the probable causes of such imperfections will be found in Chapter XXVI.

TABLE VIII

THERMAL PROPERTIES OF GASES

Name of Gas.	Chemical Symbol.	Density.*	R	r	C_p	C_v
Air	....	0.080710	53.338	1.4068	0.23727	0.16867
Acetylene	C_2H_2	0.07251	59.37			
Argon	A	0.11126	38.70	1.6667	0.24890	0.14934
Carbon dioxide †	CO_2	0.12262	35.11	1.315	0.1886	0.1434
Carbon monoxide	CO	0.07807	55.14	1.415	0.2425	0.1716
Ethylene †	C_2H_4	0.07809	55.13	1.213	0.4040	0.3331
Helium	He	0.01189	362.0	1.6667	2.3280	1.3968
Hydrogen	H_2	0.005588	770.4	1.4102	3.40559	2.41474
Marsh-gas	CH_4	0.04464	96.44	1.2646	0.59295	0.46892
Nirtic oxide	NO	0.0838	51.37	1.3994	0.23150	0.16543
Ntirogen	N_2	0.07831	55.10	1.4099	0.24356	0.17268
Oxygen	O_2	0.08921	48.256	1.3997	0.21729	0.15523
Steam †	H_2O	....	85.72	1.285	0.4614	0.3512
Sulphurous anhydride †	SO_2	0.1793	24.09	1.251	0.1544	0.1234

* Lbs. per cu.ft. at 32° F. and 1 atmosphere.
† The properties of these gases vary greatly with the temperature and pressure.

PROBLEMS

1. A quantity of air is confined within a volume of 2 cu.ft., at a pressure of 3000 lbs. per square foot. What will be its volume at the same temperature, if its pressure is reduced to 1000 lbs. per square foot? Ans. 6 cu.ft.

2. What will be the pressure of the above mass of air, if its volume is made 4 cu.ft.? Ans. 1500 lbs. per square foot.

3. A certain substance has a temperature of 50° F. What is its absolute temperature in Fahrenheit degrees? Ans. 509.6°.

4. A certain substance has an absolute temperature of 660°. What is its Fahrenheit temperature? Ans. 199.4°.

5. A certain substance has a temperature of 104° F. What is its absolute temperature in Centigrade degrees? Ans. 313.1°.

6. A gas is confined within a given volume at a temperature of 1000° absolute and has a pressure of 1000 lbs. per square foot. What will be its pressure if its temperature is raised to 1500° absolute without changing its volume?

Ans. 1500 lbs. per square foot.

7. A quantity of gas having a temperature of 600° absolute is heated until its pressure is doubled, without change in volume. What is its final temperature?

Ans. 1200° absolute, or 740.4° F.

8. A quantity of gas having a pressure of 2500 lbs. per square foot and a temperature of 60° F. is raised in temperature to 250° F. without change in volume. Find its final pressure. Ans. 3414 lbs. per square foot.

9. A quantity of gas having an initial pressure of 50 lbs. per square inch absolute, and temperature of 60° F. is raised to a pressure of 80 lbs. per square inch by heating without change in volume. Find the final temperature. Ans. 371.8° F.

10. What will be the value of the constant R for chlorine gas, chlorine gas being diatomic and the atomic weight of chlorine being 35.5? Ans. $R=21.75$.

11. A certain mixture of gases has a density of 0.88 as compared with air; what will be the value of the constant R for this mixture? Ans. $R=60.61$

12. How many pounds of the above mixture will be contained in a cylindrical gasometer 50 ft. in diameter and 40 ft. high, at atmospheric pressure and a temperature of 80° F.? Ans. 5088 lbs.

13. What volume will 1 lb. of marsh-gas (CH_4) occupy at a temperature of 100° F. and a pressure of 10,000 lbs. per square foot? Ans. 5.40 cu.ft.

14. A tank contains 2 lbs. of oxygen at a pressure of 400 lbs. per square inch absolute and a temperature of 60° F. What is its volume? Ans. 0.8707 cu.ft.

15. A cylindrical tank 6 ins. in diameter and 3 ft. long contains $1\frac{1}{4}$ lbs. of acetylene, whose formula is C_2H_2. At what temperature will the pressure within this cylinder reach 500 lbs. per square inch gage? Ans. 129.2° F.

16. A quantity of gas has a pressure of 800 mm. of mercury, a volume of 1 liter, and its temperature is 15° C. What will be its temperature when the pressure becomes 1000 mm. of mercury and the volume 1.1 liters? Ans. 123° C.

17. The specific heat of a gas at constant pressure is 3.000. The value of R for this gas is 700. Find the value of its specific heat at constant volume.

Ans. $C_v=2.101$.

18. Find the value of the constant γ for the above gas. Ans. $\gamma=1.428$.

19. Assuming that γ for air equals 1.406, compute from the value of R given in Art. 30, the value of the dynamic specific heat of air at constant volume.

Ans. $K_v=131.4$ ft.-lbs.

20. Compute the specific heat of air at constant pressure. Ans. $C_p=0.2375$.

21. What quantity of heat will be required to raise the temperature of 2 lbs. of air confined at constant pressure, from 40° to 90° F? Ans. 23.75 B.T.U.

22. What will be the increase in the intrinsic energy of the air in the above case?

Ans. 16.89 B.T.U.

23. What will be the external work performed in the above case?

Ans. 5334 ft.-lbs.

(Note the relation between the intrinsic energy, the external work, and the heat imparted.)

CHAPTER III

THE EXPANSION OF GASES

37. The Pressure Volume Diagram. The relation between the pressure and volume of a mass of homogeneous fluid expanding by any definite law may be expressed by means of an equation involving these two quantities only as variables. The locus of the simultaneous values of the pressure and volume of an expanding fluid, when plotted as a curve whose ordinates are pressures and whose abscissæ are volumes, is known as the pressure volume or PV curve of the expanding fluid. The equation of this curve is obviously the equation relating the pressure and the volume of the expanding fluid. If a series of such curves is collected on one diagram, in order to show graphically the changing states of a mass of fluid, the diagram is termed a Watt diagram, and is also known as a PV diagram or an indicator card. Since areas on such diagrams represent the product of pressure and volume, they also represent work or energy. The use of such curves and diagrams is not only convenient, but necessary, in illustrating the principles of thermodynamics, and in investigating the performance of most thermodynamic machinery.

38. The Four Cases of Expansion. When a gas is allowed to expand there are four definite conditions under which the expansion may take place. In the first case the gas expands at constant temperature, and since it does work by its expansion, this necessitates the addition of heat to the gas in order to maintain its temperature at a constant value. This is termed **isothermal** expansion. In the second case the gas expands without the addition or abstraction of heat. Since the gas does work by its expansion, some of the energy contained in the gas as heat is lost, or rather transformed, in order to perform this work. This is termed **adiabatic** expansion. In the third place, the gas expands without undergoing a change of pressure. Since it does work by its expansion, and the product of its pressure and volume (and therefore its temperature) increases, heat must be added both to do the work and to raise its temperature. This is termed **isobaric** expansion. In the fourth case the loss in intrinsic energy of the expanding gas is proportional to the external work of expansion. This is termed **polytropic** expansion.

39. Isothermal Expansion. During the isothermal expansion of a sensibly perfect gas the mass of gas being at a constant temperature, its pressure and volume must be such that $PV = WRT$, where W, R, and T are all constants. Therefore, the relation between the pressure and volume at any instant during isothermal expansion, may be expressed by the equation

$$P V = C = P_1 V_1,$$

where P and V are both variables, and C is a constant equal to the product of the initial pressure and volume. This is the equation of a second degree curve, which is known in analytic geometry as the equilateral hyperbola, the form of which is shown in Fig. 4. The curve is asymptotic to both its axes.

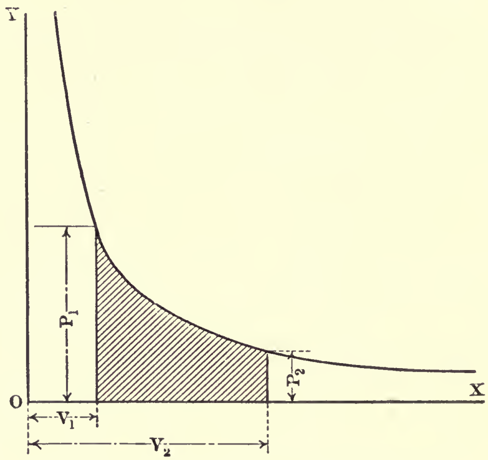


FIG. 4.—The isothermal expansion line.

40. Graphical Construction of the Equilateral Hyperbola. The isothermal expansion line may be constructed by the method shown in Fig. 5. Let A be any point on the curve for which the pressure and corresponding volume are known. Through A draw vertical and horizontal lines marked VV and HH respectively, in the figure. From O, the origin of axes, draw lines OB, OC, OD, etc. Through the intersections of these lines with VV draw horizontals as GJ, and through their intersections with HH erect perpendiculars as IJ. The intersections of these perpendiculars with the corresponding horizontals are points on the equilateral hyperbola passing through A and asymptotic to the axes. Conversely, if through two points on such an expansion line perpendiculars and horizontals are drawn, and a diagonal is drawn

through the rectangle so formed, the diagonal produced will pass through the origin of axes.

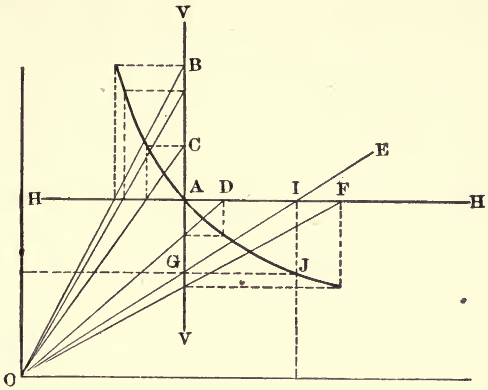


FIG. 5.—Graphical construction of the equilateral hyperbola.

41. Work Done During Isothermal Expansion. In order to find the work done by a mass of gas expanding isothermally, we must multiply the change in volume by the mean pressure during the expansion. Let us suppose that the gas expands from the condition P_1V_1, to the condition P_2V_2. The change in volume will, of course, be $V_2 - V_1$. The mean pressure during expansion is the mean ordinate included under the pressure volume-curve between the pressures P_1 and P_2. The mean ordinate multiplied by the change in volume must, of course, be equal to the work done, and therefore to the shaded area included under the curve in Fig. 4.

Since the temperature remains constant during the expansion, we have for the value of PV at any point during the expansion,

$$P\,V = C = P_1V_1 \quad \text{from whence} \quad P = \frac{C}{V}$$

Now, since the work done during any small expansion is given by the equation

$$dU = P\,dV, \qquad (1)$$

the work done during any isothermal expansion may be expressed by the equation,

$$U = \int_{V_1}^{V_2} P\,dV = \int_{V_1}^{V_2} \frac{C}{V}\,dV. \qquad (2)$$

Integrating, we have

$$U = C\left(\log_e \frac{V_2}{V_1}\right). \qquad (3)$$

Now, since the product of the pressure and volume are always equal to C, the product of the initial pressure and the initial volume will, of course, be equal to C, and we may substitute P_1V_1 for C. The ratio of the final to the initial volume is known as the ratio of expansion and usually denoted by the symbol r. We may therefore write the above equation in the form

$$U = P_1V_1 \log_e r. \quad \quad (4)$$

This may also be written in the form

$$U = W\ R\ T \log_e r. \quad \quad (5)$$

In the above equations

U = the work done, in foot-pounds, by a mass of gas expanding isothermally;
W = the mass of the gas in pounds;
T = the absolute temperature of the gas;
P_1 = the initial pressure in pounds per square foot,
V_1 = the initial volume in cubic feet; and
r = the final divided by the initial volume.

It will be noted that as the gas continues to expand to a larger and larger volume, the ratio of expansion will increase indefinitely with the volume. Hence the work done by a mass of gas expanding isothermally to an infinite volume is infinite in amount. Since the temperature of the gas remains constant, the intrinsic energy of the gas will also remain constant and the heat added to effect isothermal expansion is entirely transformed into work. Were it possible to devise an apparatus which could make use of the infinite isothermal expansion of a gas, we would have a heat engine capable of transforming the entire quantity of heat energy transferred to it, into work. However, it is, so far as we know, not possible to construct such an engine.

42. Relation between the Changes in Pressure, Volume, and Temperature of an Expanding Gas. Assume that 1 pound of a sensibly perfect gas expands by any law whatever, and thereby undergoes simultaneously a change of temperature, pressure, and volume. Let the initial temperature of the gas be T, the initial pressure P, and the initial volume V. Let us assume that the change of temperature resulting from the expansion is dT, the change of pressure is dP and the change of volume is dV. In case the pressure, temperature, or volume are increased, the respective changes will be positive, and in case they are decreased, they will be negative. Then, after expansion, we will have for the final volume $V+dV$, for the final pressure $P+dP$, and for

the final temperature $T+dT$, where dP and dT may be either positive or negative. From the characteristic equation of gases we have $PV=RT$ for 1 pound of any gas, and therefore after the expansion

$$(P+dP)(V+dV)=R(T+dT). \quad \ldots \ldots \quad (1)$$

Multiplying this out, we will have

$$PV+P\,dV+V\,dP+dP\,dV=RT+R\,dT. \quad \ldots \ldots \quad (2)$$

Since $PV=RT$ and in the limit $dP\,dV$ vanishes, we may write

$$P\,dV+V\,dP=R\,dT \quad \ldots \ldots \quad (3)$$

or

$$dT=\frac{P\,dV+V\,dP}{R}. \quad \ldots \ldots \quad (4)$$

which is an equation relating the change in temperature of an expanding gas to the changes in pressure and volume. If the temperature falls, dT will be negative, and since in most cases of expansion the pressure also falls, dP is also usually negative.

43. Adiabatic Expansion. When a gas expands adiabatically, it loses heat, since a part of its intrinsic energy is transformed into the mechanical work of expansion. If the expansion be infinitesimal, the work done is expressed by the equation

$$dU=P\,dV, \quad \ldots \ldots \quad (1)$$

in which dU is the quantity of work performed by the gas, P is the pressure and dV is its change in volume. The loss of intrinsic energy is equal to the specific heat of the gas at constant volume, K_v, multiplied by the change in temperature, dT. Since the temperature falls, dT is negative, and therefore the heat lost by the gas is expressed by the equation

$$dH=-K_v\,dT. \quad \ldots \ldots \quad (2)$$

Since the loss of intrinsic energy is equal to the work of expansion, we may write

$$P\,dV=-K_v\,dT. \quad \ldots \ldots \quad (3)$$

We have shown in the preceding paragraph that

$$dT=\frac{P\,dV+V\,dP}{R}. \quad \ldots \ldots \quad (4)$$

Substituting (4) in (3), we have

$$P\,dV = -\frac{K_v}{R}(V\,dP + P\,dV). \qquad (5)$$

Since

$$K_v = K_p - R, \quad \text{and} \quad \frac{K_p}{K_v} = \gamma,$$

we have

$$\frac{K_v}{R} = \frac{1}{\gamma - 1}. \qquad (6)$$

Substituting (6) in (5) we have

$$P\,dV = -\frac{1}{\gamma - 1}(V\,dP + P\,dV), \qquad (7)$$

which becomes

$$\gamma P\,dV = -V\,dP. \qquad (8)$$

Dividing through by PV we have

$$\gamma\frac{dV}{V} = -\frac{dP}{P}. \qquad (9)$$

Integrating each side between corresponding limits

$$\gamma\int_{V_1}^{V}\frac{dV}{V} = \int_{P_1}^{P}-\frac{dP}{P}, \qquad (10)$$

which becomes

$$\gamma\,(\log_e V - \log_e V_1) = \log_e P_1 - \log_e P, \qquad (11)$$

or

$$\gamma \log_e \frac{V}{V_1} = \log_e \frac{P_1}{P}, \qquad (12)$$

whence

$$\left(\frac{V}{V_1}\right)^{\gamma} = \frac{P_1}{P}, \qquad (13)$$

and

$$P\,V^{\gamma} = P_1 V_1^{\gamma}. \qquad (14)$$

In the above equation, P is the pressure and V is the corresponding volume of a quantity of sensibly perfect gas undergoing adiabatic expansion (or compression) when the original pressure of the gas was P_1 and its original volume V_1. This expression may also be written

$$P\,V^{\gamma} = C, \qquad (15)$$

in which C is a constant.

44. Relation between Initial and Final Pressure, Volume, and Temperature of a Gas Expanding Adiabatically. If the gas be assumed to have expanded to some pressure P_2, and corresponding volume V_2 we will of course have the relation

$$P_2 V_2^r = P_1 V_1^r. \quad \quad (1)$$

This may be written

$$\left(\frac{V_2}{V_1}\right)^r = \frac{P_1}{P_2}. \quad \quad (2)$$

For the characteristic equation of gases, we have for 1 pound of any gas

$$T = \frac{P\,V}{R},$$

and therefore

$$\frac{T_1}{T_2} = \frac{P_1 V_1}{P_2 V_2} = \left(\frac{P_1}{P_2}\right)\left(\frac{V_1}{V_2}\right). \quad \quad (3)$$

For $\frac{P_1}{P_2}$ in (3) we may substitute its value from equation (2), and obtain the relation

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{r-1}. \quad \quad (4)$$

From number (2) we may write

$$\frac{V_2}{V_1} = \left(\frac{P_1}{P_2}\right)^{\frac{1}{r}}. \quad \quad (5)$$

Substituting this value for $\frac{V_1}{V_2}$ in equation (3) we have

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{r-1}{r}}. \quad \quad (6)$$

Solving equations (2) and (4) for V_2, (1) and (6) for P_2, and (4) and (6) for T_2 we will have

A. $V_2 = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{r}}$. B. $V_2 = V_1\left(\frac{T_1}{T_2}\right)^{\frac{1}{r-1}}$.

C. $P_2 = P_1\left(\frac{V_1}{V_2}\right)^{r}$. D. $P_2 = P_1\left(\frac{T_2}{T_1}\right)^{\frac{r}{r-1}}$.

E. $T_2 = T_1\left(\frac{V_1}{V_2}\right)^{r-1}$. F. $T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{r-1}{r}}$.

These equations may be readily solved by the use of a table of logarithms when they are written in the form

$$\text{A}'.\ \log V_2 = \log V_1 + \frac{1}{\gamma} \log \left(\frac{P_1}{P_2}\right).$$

$$\text{B}'.\ \log V_2 = \log V_1 + \frac{1}{\gamma - 1} \log \left(\frac{T_1}{T_2}\right).$$

$$\text{C}'.\ \log P_2 = \log P_1 + \gamma \log \left(\frac{V_1}{V_2}\right).$$

$$\text{D}'.\ \log P_2 = \log P_1 + \frac{\gamma}{\gamma - 1} \log \left(\frac{T_2}{T_1}\right).$$

$$\text{E}'.\ \log T_2 = \log T_1 + (\gamma - 1) \log \left(\frac{V_1}{V_2}\right).$$

$$\text{F}'.\ \log T_2 = \log T_1 + \frac{\gamma - 1}{\gamma} \log \left(\frac{P_2}{P_1}\right).$$

By means of these relations we may compute the final temperature, pressure, or volume of a mass of gas expanding adiabatically, when its initial temperature, pressure or volume is known, and also the ratio of its initial and final pressure, temperature, or volume. It is to be noted that these equations will be true, no matter what system of units be employed. For instance, the temperatures may be expressed in Centigrade or Fahrenheit degrees on the absolute scale, the pressure in pounds per square inch or per square foot, or in atmospheres, or in inches or millimeters of mercury, and the volumes in per cent, in cubic feet, in cubic inches, or in cubic centimeters or liters. So long as the same system of units is employed throughout an equation, the results obtained will be correct.

45. Work Done During Adiabatic Expansion. The amount of work done by a mass of gas expanding adiabatically will of course be equal to the heat lost by the gas. Therefore we may write

$$U = W\,K_v(T_1 - T_2). \quad \ldots\ldots\ldots \quad (1)$$

Substituting for T_1 the value $\frac{P_1V_1}{W\,R}$ and for T_2 the value $\frac{P_2V_2}{W\,R}$, we may write this expression,

$$U = W\,K_v\left(\frac{P_1V_1}{W\,R} - \frac{P_2V_2}{W\,R}\right). \quad \ldots\ldots\ldots \quad (2)$$

Clearing this, we will have

$$U=\frac{K_v}{R}(P_1V_1-P_2V_2). \qquad (3)$$

From equation (6), Art. 43, we have the relation

$$\frac{K_v}{R}=\frac{1}{\gamma-1}. \qquad (4)$$

Substituting this we will have

$$U=\frac{P_1V_1-P_2V_2}{\gamma-1}, \qquad (5)$$

in which U = the number of foot-pounds of work done by a mass of gas expanding adiabatically;
P_1 = the initial pressure of the gas in pounds per square foot;
P_2 = the final pressure in the same units;
V_1 = the initial volume of the gas in cubic feet;
V_2 = the final volume in the same units.

This same result may be obtained by the integration of the expression

$$U=\int_{V_1}^{V_2}P\,dV, \qquad (6)$$

in which we substitute for P the value obtained from equation (C), Art. 44, namely,

$$P=P_1\left(\frac{V_1}{V}\right)^{\gamma}.$$

The integral of this expression is, of course, equal to the shaded area in Fig. 6, which shows the pressure-volume curve of a mass of gas expanding adiabatically. This curve is also, like the rectangular hyperbola, asymptotic to both axes, but it will be noted that the area included under this curve from the volume V_1 to an infinite volume is not infinite in amount, but is a definite and finite quantity, as will be seen from equation (1) in this article. By such an expansion the gas will part with the entire amount of intrinsic energy which it contains, and the work of expansion is limited to this quantity of energy.

46. Graphical Construction of the Curve whose Equation is $PV^n = C$. Any curve represented by the general equation

$$PV^n = C$$

may be constructed graphically by the method shown in Fig. 7, when one point on the curve, and the value of the index n are known. Let the coordinates of point M be P_1 and V_1, and the coordinate axes be

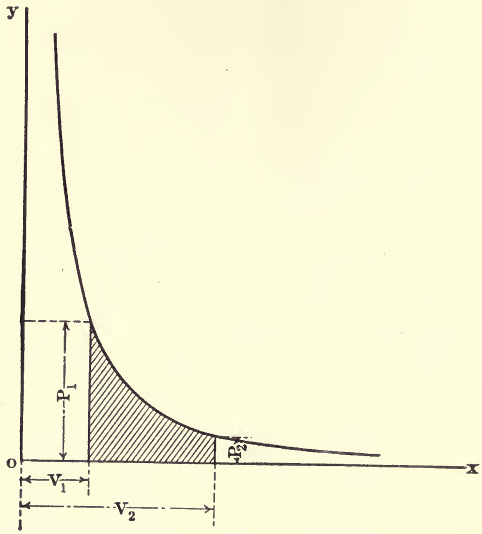


FIG. 6.—The adiabatic expansion line.

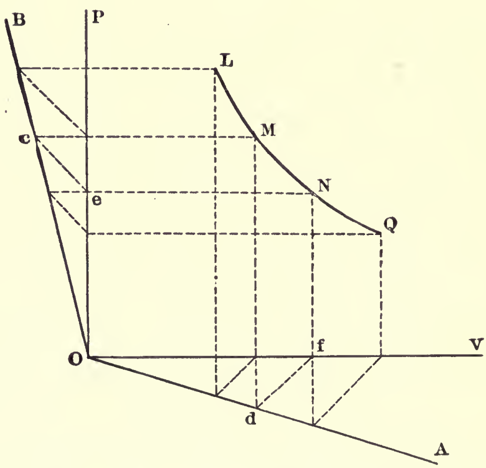


FIG. 7.—Graphical construction of the curve $P\,V^n = K$.

OP and OV. Draw AO, making any convenient angle AOV with OV. Then let

$$X = (1 + \tan AOV)^n.$$

Determine the angle BOP, whose tangent is given by the equation,

$$\tan BOP = \frac{X-1}{X},$$

and construct angle BOP. Through M draw the horizontal line Mc and the vertical line Md, intersecting OB at c and Oa at d respectively. Through c and d draw ce and df, making angles of 45° with the coordinate axes, and intersecting them at e and f. Through e draw a horizontal and through f a perpendicular, intersecting at N, which will be a second point on the curve. In like manner points l and q and as many more points as are desired, may be located.

An inspection of the figure will show that if P_1 and V_1 be the coordinates of M, and P and V those of N, that

$$P = P_1 - P_1 \tan BOP = P_1(1 - \tan BOP), \quad \ldots \quad (1)$$

and

$$V = V_1 + V_1 \tan AOV = V_1(1 + \tan AOV). \quad \ldots \quad (2)$$

From equation (2) we may write

$$V^n = V_1^n(1 + \tan AOV)^n. \quad \ldots \quad (3)$$

Hence

$$P\,V^n = P_1V_1^n = P_1V_1^n(1 - \tan BOP)(1 + \tan AOV)^n. \quad \ldots \quad (4)$$

Dividing both sides by $P_1V_1^n$ we have

$$(1 - \tan BOP)(1 + \tan AOV)^n = 1. \quad \ldots \quad (5)$$

From which we deduce that

$$(1 + \tan AOV)^n = X = \frac{1}{1 - \tan BOP}. \quad \ldots \quad (6)$$

Clearing (6),

$$X - X \tan BOP = 1. \quad \ldots \quad (7)$$

Solving for $\tan BOP$,

$$\tan BOP = \frac{X-1}{X}. \quad \ldots \quad (8)$$

Hence a curve constructed in the manner described will satisfy the equation

$$P\,V^n = P_1V_1^n = C.$$

It will often be more convenient to determine the angles AOV and BOP by computing the coordinates of a second point upon the curve, such as N, and then after drawing Ne and Nf, ec, and fd, and Mc and Md, finally draw OB and OA through the intersections of ec with Mc, and fd with Md, respectively.

47. Isobaric Expansion. The equation for isobaric expansion is $P = k$, a constant, since the pressure remains constant. The PV curve is therefore a horizontal line.

During isobaric expansion, the quantity of heat added to the gas is of course,

$$W\,K_p(T_2 - T_1).$$

The work done during isobaric expansion is equal to

$$P(V_2 - V_1).$$

The amount of intrinsic energy imparted to the gas is equal to

$$W K_v(T_2 - T_1).$$

We may also express the amount of work done during the isobaric expansion in terms of the temperatures by the expression

$$W R(T_2 - T_1).$$

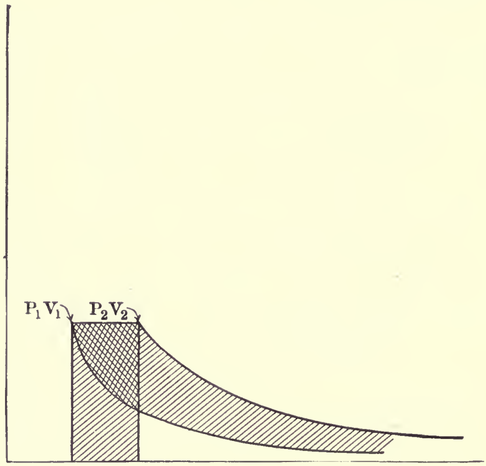


FIG. 8.—Work of isobaric expansion and increase in intrinsic energy.

We may express the quantity of heat imparted to the gas in terms of the pressure and volumes, by the expression

$$\frac{\gamma P(V_2 - V_1)}{W(\gamma - 1)}.$$

If an adiabatic be drawn from the state P_1V_1 and another to be drawn from the state P_2V_2 of a gas undergoing isobaric expansion, as in Fig. 8, the intrinsic energy of the gas at the beginning of expansion will be represented by the area under the first adiabatic, the intrinsic energy at the end of the expansion will be represented by the area under the second adiabatic, and the energy imparted to the gas will be represented by the difference between these areas plus the work of expansion which is represented by the area under the isobaric line. This is, obviously, the shaded area plus twice the blackened area.

48. Polytropic Expansion. The work performed by a mass of gas undergoing polytropic expansion is equal to a constant times the loss in intrinsic energy. Hence we may write

$$P\,dV = -a\,K_v\,dT, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

by analogy with equation (3), Art. 43. Making the same substitutions as were made in the former equation and transposing the constant a to the left-hand member we will obtain

$$\left(\frac{\gamma - 1 + a}{a}\right)\frac{dV}{V} = \frac{-dP}{P}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

which is similar in form to equation (9) of that article. Substituting n for $\frac{\gamma-1+a}{a}$, we will finally obtain the relation,

$$P\,V^n = P_1 V_1{}^n = C, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

which is similar in form to equation (14) of Art. 43, and is the equation giving the relation between the pressure and volume of a mass of gas undergoing polytropic expansion.

49. The Relation between the Initial and Final Pressure, Volume, and Temperature of a Gas Undergoing Polytropic Expansion. From equation (e) in the preceding paragraph, the relations between the initial and final pressure temperature and volume of a mass of gas undergoing polytropic expansion may be deduced by the methods outlined in Art. 44. By substituting n for γ in the equations A to F and A' to F' in that article, the relation between the initial and final pressure, temperature, and volume of a gas undergoing polytropic expansion, may be computed. In the same way by substituting aK_v for K_v and n for γ in the equations in Art. 45, the work done during polytropic expansion may be computed. In case the value of the exponent n is very nearly unity, exact computations of the work done during polytropic expansion are not feasible. In case it is desirable to determine exactly the work done during such expansion the area included under the expansion line may be measured, and the work of expansion computed from the measured area and the known scales of the diagram.

50. Special Cases of Polytropic Expansion. It may be noted that the isothermal, adiabatic, and isobaric expansion may all be considered special cases of polytropic expansion. When $a = \infty$,

$$n = \frac{\gamma - 1 + \infty}{\infty} = 1,$$

which is the case of isothermal expansion where there is no change in intrinsic energy. When $a=1$,

$$n=\frac{\gamma-1+1}{1}=\gamma,$$

which is the case of adiabatic expansion, where the change in intrinsic energy is equal to the work performed. When $a=1-\gamma$, $n=0$, which is the case of isobaric expansion, where the pressure is constant. When $a=0$,

$$n=\frac{\gamma-1+0}{0}=\infty,$$

which is the case of change of pressure without change of volume (i.e., change in intrinsic energy without performance of work).

51. Expansion in Conducting Cylinders. Gases do not remain in thermodynamic equilibrium while they are being expanded or compressed in practical thermodynamic machines, since there will in general be a difference in temperature between the expanding gas and the conducting walls of the containing vessel. As a result the temperature of the layers of gas close to the walls will be different from that of the mass of the gas. It is found, however, that under these conditions the pressure-volume curve of expansion or compression is very nearly a line of polytropic expansion, and the value of the index lies between 1 and γ. Such expansion or compression may be treated as though it were polytropic, the value of the index n in the equation $PV^n=C$ being determined from the actual expansion curve by the equation

$$n=\frac{\log\frac{P}{P_2}}{\log\frac{V_2}{V_1}},$$

in which P_1 and V_1 and P_2 and V_2 are corresponding absolute pressures and volumes as derived from the actual expansion line taken from an indicator card.

52. Compression the Converse of Expansion. The process of compression is the reverse of the process of expansion, the volume of the gas progressively diminishing as the process continues. In order to compress a gas, work must be done upon it, which accounts for the fact that if proper substitutions be made in any of the formulæ for the work done by an expanding gas, we will, in the case of compression, get a negative answer. For instance, in the case of isothermal expansion, the final volume is less than the initial volume, the ratio of the volumes is less than unity and the logarithm of the ratio is a negative quantity (see Art. 41, equation (4)), indicating that the gas does negative work during compression. If proper substitutions be made in the equations giving quantities of heat absorbed or rejected by gases undergoing compression, the answers will also be negative, indicating that in the case of compression, the heat transfer is in the opposite direction to what it is in the case of expansion.

53. The Velocity of Sound. If the pressure of a mass of gas be suddenly increased at some point, the pressure of the entire mass is not raised instantly, but

the increase of pressure travels from point to point in the gas with a velocity depending on the nature and temperature of the gas. Assume a column of gas whose cross-section is 1 square foot and whose length is indefinite, to be confined within a tube, under a pressure of P pounds per square foot and at the temperature T. If the pressure at one end of this tube be increased suddenly by applying the force dP to the piston shown in Fig. 9 for one second, the increase in pressure, dP, will be transmitted to the right with the velocity V feet per second, and at the end of one second V cubic feet of gas will be compressed. The mass of this gas will be

$$W = \frac{PV}{RT}. \qquad (1)$$

Since the compression of the gas is sudden, the gas does not have time to part with its heat to surrounding objects, and the compression is adiabatic, the relation between the pressure and volume being expressed by the equation

$$PV^{\gamma} = C \qquad (2)$$

from whence

$$V^{\gamma} = CP^{-1}. \qquad (3)$$

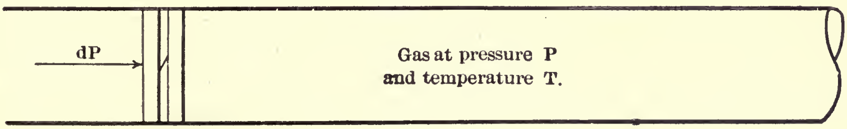


FIG. 9.

Differentiating this expression we have

$$\gamma V^{\gamma-1} dV = -CP^{-2} dP, \qquad (4)$$

or

$$dV = -\frac{C\,dP}{\gamma V^{\gamma-1} P^2}. \qquad (5)$$

We may substitute for C its value from equation (2) and obtain

$$dV = -\frac{V\,dP}{\gamma P}. \qquad (6)$$

In the above expression dV is the change in volume of the gas, dP is the change in pressure, and P and V are the initial pressure and volume. It will be noted that if the pressure increases, the volume diminishes, as is indicated by the minus sign. As a result of the application of the constant force dP to the gas, the end of the column is moved, in one second, a distance dV, and the center of gravity of the column is moved in the same time a distance $\frac{dV}{2}$. The acceleration produced by this constant force in the column of gas is twice the distance which the gas was moved in the first second, or dV feet per second per second. Now, from a well-known principle in dynamics, namely, force = mass × acceleration, we have

$$dP = -\frac{W}{g} dV. \qquad (7)$$

Substituting from 1 and 6,

$$dP=\frac{P\,V}{g\,R\,T}\left(\frac{V\,dP}{\gamma\,P}\right). \quad \ldots \ldots \ldots \quad (8)$$

Solving for V we will have

$$V=\sqrt{\gamma\,g\,R\,T}, \quad \ldots \ldots \ldots \quad (9)$$

in which V is the rate of transmission of pressure in gas in feet per second, and T is the absolute temperature of the gas in Fahrenheit degrees.

54. Velocity of Transmission of Explosive Pressure in Confined Gases. The above equations will be true only when the increase in pressure is infinitesimal as compared with the actual pressure of the gas. This is true in the case of sound waves, which are transmitted in the gas with the velocity given by the above equation. When, however, the wave is caused by a violent explosion which produces a large increase in pressure, the velocity of transmission of the pressure is higher than the above equation would indicate.

If it be assumed that the increase in pressure produced by the suddenly applied force is large, the final pressure produced may be represented by P_2 while the initial pressure may be represented by P_1. The final volume will be, from equation (A), Art. 44,

$$V_2=V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{\gamma}}. \quad \ldots \ldots \ldots \quad (8)$$

The decrease of the volume of the mass, and therefore, as has already been shown, the acceleration of the mass, will be

$$V_1-V_2=V_1\left(1-\left(\frac{P_1}{P_2}\right)^{\frac{1}{\gamma}}\right). \quad \ldots \ldots \ldots \quad (9)$$

We may therefore write, by analogy from equation (8) of the previous article,

$$P_2-P_1=\frac{P_1V_1^2}{g\,R\,T}\left(1-\left(\frac{P_1}{P_2}\right)^{\frac{1}{\gamma}}\right). \quad \ldots \ldots \ldots \quad (10)$$

Solving for V_1 we will have

$$V_1=\sqrt{\frac{g\,R\,T\,(P_2-P_1)}{P_1\left(1-\left(\frac{P_1}{P_2}\right)^{\frac{1}{\gamma}}\right)}}, \quad \ldots \ldots \ldots \quad (11)$$

which is an expression giving the velocity of transmission of pressure in a gas when the increase in pressure is great as compared with the original pressure.

When a combustible mixture of gases is confined under pressure and ignited at some point, as is the case in most types of gas engines, the increase in pressure produced by the local explosion is transmitted throughout the mass of the gas in the same manner as the pressure was transmitted through the column described in Art. 53. As the pressure wave proceeds through the gas, it compresses it adiabatically. If the initial temperature of the gas is sufficiently high, so that the adiabatic compression heats the mixture to its kindling point, the flame of combustion will proceed through the mass with the velocity indicated by equation (11) of the preceding paragraph. If, however, the temperature of the gas is not sufficiently high for this action to take place, the pressure in the gas will increase gradually, the flame being propagated from point to point by heat conduction and radiation, with

a velocity many hundred times lower than that given. The phenomena of pressure transmission are of great practical importance in the theory of gas engines and gaseous explosions.

55. Theory of the Flow of Gas through an Orifice. When a gas flows through a nozzle, it will be found that as each particle of the gas passes through it will expand in volume and increase in velocity. If the ratio of expansion in volume as the gas passes from one cross-section of the nozzle to another is less than the ratio of increase in velocity, it must follow that the nozzle is less in cross-section at the second point than at the first, the nozzle being convergent between the two sections. If, on the other hand, the ratio of expansion is greater than the ratio of increase in velocity, the cross-section of the nozzle will be greater at the second point than at the first, the nozzle being divergent between the two sections.

In passing through a nozzle, a gas will of course neither gain nor lose heat, on account of the small time which each particle takes in passing through. This being the case, the kinetic energy of the quantity of gas which passes a given cross-section of the nozzle in a given time, plus the work it does in displacing the gas in the region

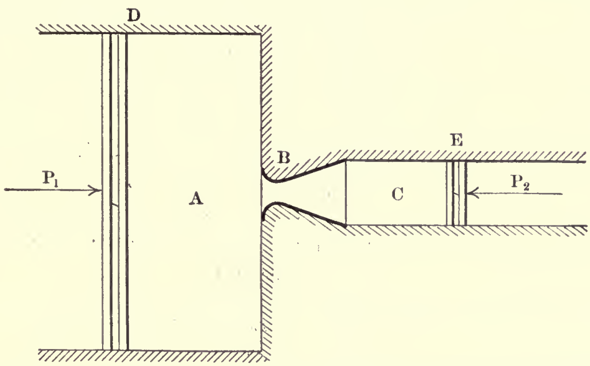


Fig. 10.—Ideal apparatus illustrating the flow of gas through a nozzle

into which it rushes, must be equal to the loss in intrinsic energy of the gas, plus the work done upon it by the advancing mass of gas which takes its place in the region from which the gas flows.

56. Flow through a Nozzle. This will be apparent from a consideration of Fig. 10, in which A is a cylinder and B a nozzle. The cross-section of the nozzle is very small in comparison with that of the cylinder, so that the velocity of the gas in the cylinder may be neglected. The gas emerging from the nozzle passes into the tube C, whose cross-section is the same as that of the nozzle at the point where the nozzle terminates. Assume that cylinder A is filled from point D with gas having a pressure P_1 and a temperature T_1 and the tube is filled to the point E with gas having a pressure P_2 and a temperature T_2. At E in the tube and at D in the cylinder are pistons, which, of course, exert upon the gas a pressure equal to the pressure exerted upon them by the gas. Since the pressure in tube C is less than the pressure in cylinder A, the gas will flow from A to C through the nozzle, and if the pressure in A and C are to remain constant, the pistons must both move to the right. If a certain quantity of gas be supposed to flow from A to C in one second, then its volume in A may be assumed to be V_1 and its volume in C may be assumed to be V_2. The amount of work done by piston D upon the gas during this second

is equal to P_1V_1, and the amount of work done by the gas upon the piston E is P_2V_2. The intrinsic energy of the gas has been diminished by the amount

$$W\,K_v(T_1-T_2)$$

and the kinetic energy gained by the gas is equal to

$$\frac{W\,v^2}{2\,g},$$

in which W is the weight of gas which flows through the nozzle in one second, and v is the velocity of the gas flowing into tube C.

57. Determination of the State of the Gas Passing the Throat of a Nozzle. At first the gain in velocity as the gas passes successive sections of the nozzle will proceed at a greater rate than the increase in the volume of the mass, and the successive sections will diminish in area, the nozzle being convergent. When a certain point is reached, however, the rate of gain in volume increases more rapidly than the rate of gain in velocity, and the nozzle from that point outward must be made divergent. The point of minimum cross-section is known as the throat of the nozzle, and the quantity of gas which the nozzle will pass will obviously depend upon the area of this cross-section of the nozzle. Let v be the velocity of the gas passing this cross-section, let W be the number of pounds passing this cross-section per second, let T be the absolute temperature of the gas passing this cross-section, let P be the pressure of the gas passing this cross-section, and let T_1 and P_1 be the temperature and pressure of the gas entering the nozzle. Then we will have, collecting and equating the terms given at the end of Art. 56,

$$W\,K_v(T_1-T)+P_1V_1=\frac{W\,v^2}{2\,g}+P\,V. \qquad (1)$$

This may be written

$$W\,K_v(T_1-T)+W\,R\,T_1=\frac{W\,v^2}{2\,g}+W\,R\,T, \qquad (2)$$

collecting, we will have

$$(K_v+R)(T_1-T)=\frac{v^2}{2\,g}, \qquad (3)$$

or

$$K_p(T_1-T)=\frac{v^2}{2\,g}, \qquad (4)$$

solving

$$v=\sqrt{2\,g\,K_p(T_1-T)}=k_1\sqrt{T_1-T}. \qquad (5)$$

Also if V = the volume of W pounds of gas of temperature T and pressure P, then

$$V=\frac{W\,R\,T}{P}=k_{11}\frac{T}{P}. \qquad (6)$$

The area of the nozzle at any cross-section is $\frac{V}{v}$, and therefore we may write

$$a=\frac{k\,T}{P\sqrt{T_1-T}}. \qquad (7)$$

The area a will therefore be a minimum when

$$\frac{P\sqrt{T_1-T}}{T}\ \text{is a maximum}. \qquad (8)$$

From Art. 44, equation (D),

$$P=P_1\left(\frac{T}{T_1}\right)^{\frac{\gamma}{\gamma-1}}. \qquad (9)$$

Substituting this in (8) we will have the expression

$$\frac{P_1 T^{\frac{\gamma}{\gamma-1}}\sqrt{T_1-T}}{T\, T_1^{\frac{\gamma}{\gamma-1}}}. \qquad (10)$$

Since P_1 and T_1 are constants, this will be a maximum when

$$T^{\frac{1}{\gamma-1}}\sqrt{T_1-T} \text{ is a maximum.} \qquad (11)$$

Squaring, we have the expression,

$$T_1 T^{\frac{2}{\gamma-1}}-T^{\frac{\gamma+1}{\gamma-1}}. \qquad (12)$$

Differentiating and equating to zero, we have

$$\frac{2}{\gamma-1}\,T_1\,T^{\frac{3-\gamma}{\gamma-1}}-\frac{\gamma+1}{\gamma-1}\;T^{\frac{2}{\gamma-1}}=0. \qquad (13)$$

Solving for T_1,

$$\frac{2}{\gamma-1}T_1=\left(\frac{\gamma+1}{\gamma-1}\right)\frac{T^{\frac{2}{\gamma-1}}}{T^{\frac{3-\gamma}{\gamma-1}}}, \qquad (14)$$

from whence solving for T,

$$T=\frac{2}{\gamma+1}T_1, \qquad (15)$$

in which T is the absolute temperature of the gas passing the throat or minimum section of the nozzle. Now

$$P=P_1\left(\frac{T}{T_1}\right)^{\frac{\gamma}{\gamma-1}}. \qquad (16)$$

Substituting (15) in (16) we have,

$$P=P_1\left(\frac{\frac{2}{\gamma+1}T_1}{T_1}\right)^{\frac{\gamma}{\gamma-1}}=P_1\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}, \qquad (17)$$

in which P is the absolute pressure of the gas passing the throat of the nozzle, when the pressure in the region beyond the throat is equal to or less than the value of P given by the above equation. If the pressure in the region beyond is greater than this value, the nozzle is not of the proper form, and our conclusions do not hold. From (5), the velocity of the gas passing the throat is

$$v=\sqrt{2gK_p(T_1-T)}=\sqrt{2gK_p\frac{(\gamma-1)}{(\gamma+1)}T_1}, \qquad (18)$$

which is a fixed quantity, if the pressure of the region into which the gas escapes is less than

$$P_1\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}.$$

Now the volume of gas passing the throat in one second is, of course,

$$V = a\,v = a\sqrt{2g\,K_p\left(\frac{\gamma-1}{\gamma+1}\right)T_1}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

also

$$V = \frac{W\,R\,T}{P}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

Substituting the value of T from (15), and of P from (17) in (20), we will have

$$V = W\,R\,\frac{\frac{2}{\gamma+1}T_1}{\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}P_1} = \frac{W\,R\,T_1}{\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}P_1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

Substituting (21) in (19), and solving for W we have

$$W = \frac{a\,P_1}{R\,T_1}\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}\sqrt{2\,g\,K_p\left(\frac{\gamma-1}{\gamma+1}\right)T_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

Simplifying, this becomes

$$W = \frac{a\,P_1}{\sqrt{R\,T_1}}\sqrt{\frac{2\,g\,\gamma}{\gamma+1}}\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

which, for any particular gas, becomes

$$W = k_1\,a\,\frac{P_1}{\sqrt{T_1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

in which k_1 is a constant depending only on the physical qualities of the gas, and is found from the expression,

$$k_1 = \sqrt{\frac{2\,g\,\gamma}{R(\gamma+1)}\left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

For air we have $k_1 = .526.$

For any particular nozzle the equation becomes

$$W = C_1\frac{P_1}{\sqrt{T_1}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

58. Actual Discharge from a Nozzle. On account of friction, not all of the intrinsic energy made available by a given fall in pressure can be transformed into kinetic energy. Hence the quantity of gas discharged from an orifice per second will be less than is indicated by equation (23) in the preceding paragraph. It is probable that the friction increases somewhat with the density of the gas, and it is certain that it depends a good deal upon the form and workmanship of the nozzle. We may therefore write for the quantity of gas discharged per second through any orifice, from a region in which the pressure is P_1, into another region in which the pressure is equal to or less than

$$P_1\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}},$$

the equation

$$W = k\,a\,\frac{P_1}{\sqrt{T_1}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

in which W = the number of pounds of gas discharged per second.
a = the minimum area of the orifice in square inches;
P_1 = the absolute pressure of the gas in pounds per square inch just previous to entering the orifice;
T_1 = the absolute temperature of the gas, just previous to entering the orifice;
k = a constant depending on the nature of the gas and the form and smoothness of the orifice, but whose greatest value will be less than that of k_1 in equation (25) from the preceding paragraph.

59. Discharge against High Back Pressures. In case the pressure in the region into which the gas flows is greater than

$$P_1\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}},$$

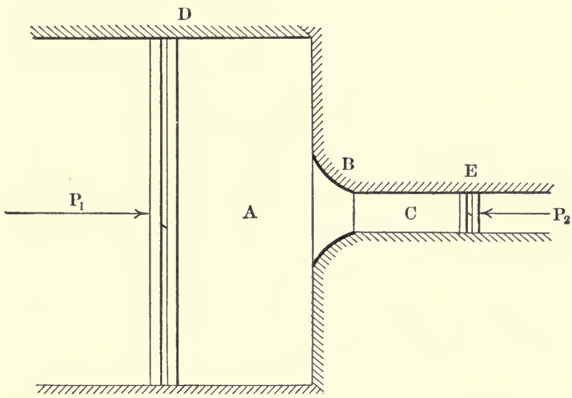


FIG. 11.—Illustrating the flow of gas through a convergent nozzle.

and the orifice has no throat, being, for instance, of the form shown in Fig. 11, then we will have as before from equation (5), Art. 57,

$$v = \sqrt{2\,g\,K_p\,(T_1 - T_2)}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

also from equation (6), Art. 57,

$$V_2 = \frac{W\,R\,T_2}{P_2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

from which we may write, since $V_2 = a\,v$,

$$\frac{W\,R\,T_2}{P_2} = a\sqrt{2\,g\,K_p\,(T_1 - T_2)}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

or

$$W = \frac{a\,P_2}{R\,T_2}\sqrt{2\,g\,K_p\,(T_1 - T_2)}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Substituting from equation (F), Art. 44, for T_2, we have,

$$W = \frac{a P_2}{V_1 P_1} \sqrt{2 g K_p T_1 \left(1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right)} \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} \quad . \; . \; . \; . \quad (5)$$

Clearing and simplifying

$$W = a \frac{P_1}{\sqrt{T_1}} \sqrt{\frac{2g}{R}} \sqrt{\frac{\gamma}{\gamma-1} \left(\frac{P_2}{P_1}\right)^{\frac{2}{\gamma}} - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma+1}{\gamma}}} \quad . \; . \; . \; . \; . \; . \; . \quad (6)$$

This expression may be written in the form

$$W = k_1 a \frac{P_1}{\sqrt{T_1}} \sqrt{r^{\left(\frac{2}{\gamma}\right)} - r^{\left(\frac{\gamma+1}{\gamma}\right)}}, \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (7)$$

in which W, a, P_1, and T_1 are as in equation (1), Art. 57,

$$r = \frac{P_2}{P_1} \quad \text{and} \quad k_1 = \sqrt{\frac{2 g \gamma}{R(\gamma-1)}}.$$

Equation (7), given above, may be transformed into

$$W = k_{11} \frac{a}{\sqrt{T_1}} \sqrt{P_2 \left(P_1^{\frac{2\gamma-2}{\gamma}} P_2^{\frac{2-\gamma}{\gamma}} - P_1^{\frac{\gamma-1}{\gamma}} P_2^{\frac{1}{\gamma}}\right)}. \quad . \; . \; . \; . \quad (8)$$

It will be noted that the sums of the exponents

$$\frac{2\gamma-2}{\gamma} + \frac{2-\gamma}{\gamma} \quad \text{and} \quad \frac{\gamma-1}{\gamma} + \frac{1}{\gamma}$$

are unity in each case. Hence if P_1 and P_2 are not widely different in value, we may write for the radical, without serious error, a simple expression which transforms equation (8) to

$$W = k a \sqrt{\frac{P_2 (P_1 - P_2)}{T_1}}, \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (9)$$

in which W, a, P_2, and P_1, and T_1 are as in equation (8), and k is a constant to be determined experimentally. Flügner gives for the value of k for air the figure 1.06. The value of this constant will, however, depend upon the character of the orifice. For any particular nozzle the equation becomes

$$W = C_2 \sqrt{\frac{P_2 (P_1 - P_2)}{T_1}}. \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (10)$$

60. Calibration of Orifices for the Measurement of Air. A method which is frequently employed for determining quantities of air is to allow the air to flow through a nozzle under known conditions. If this nozzle is of the proper form and of first-class workmanship, and its dimensions are accurately known, the quantity of air flowing may be readily determined by means of the equations which have been developed in the preceding articles. However, it is often convenient to use nozzles so small that their exact measurement is difficult, and it is then of importance

to determine the value of the constant C in equation (26), Art. 57, or equation (10), Art. 59. This may be done in the following manner:

A receiver, as shown in Fig. 12, of known volume, is filled with air or other gas at any convenient pressure, all means of egress from the receiver being stopped, except through the nozzle which is to be tested. Simultaneous readings of the pressure in the receiver, and the temperature of the air entering the nozzle, are made by means of the pressure gage and thermometer shown in the figure at regular intervals, say every fifteen seconds. The barometer reading is also noted during the experiment. A smooth curve may now be plotted showing the relation of the pressure of the gas in the receiver to the time, and another showing the relation of the temperature of the gas in the receiver to the time.

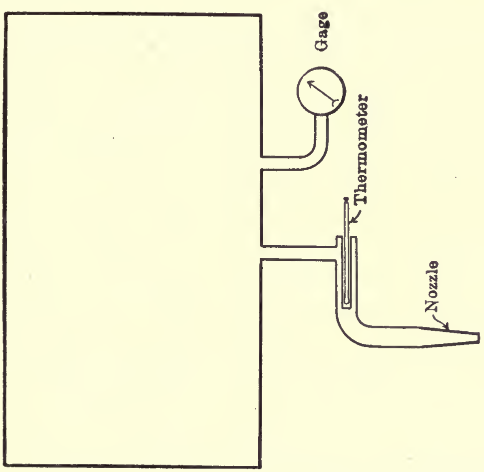


FIG. 12.—Reservoir for the calibration of air nozzles.

If a tangent be drawn to the time-pressure curve at any point we may, from its intercepts, determine the value of the differential $\frac{dP}{dt}$. In like manner, if a tangent be drawn to the time-temperature curve at the corresponding points, we may determine the differential $\frac{dT}{dt}$. During the progress of the experiment the reservoir contains a variable quantity of gas under a variable pressure and at a variable temperature, but the volume of the gas and the quantity R for the gas will of course be constant. We may write from the characteristic equation of gases,

$$P = \frac{R}{V} W T.$$

Differentiating this expression, we will have

$$dP = \frac{R}{V}(T\,dW + W\,dT).$$

Dividing this through by dt, we will have

$$\frac{dP}{dt} = \frac{R}{V}\left(T\frac{dW}{dt} + W\frac{dT}{dt}\right).$$

In this equation, we may substitute for $\frac{dT}{dt}$ and $\frac{dP}{dt}$ their values as obtained from the tangents to the two curves, for T the absolute temperature of the gas at the instant for which the tangents are drawn, and for W the mass of gas contained within the reservoir at that instant, as computed from its pressure, volume, and temperature, and solve for $\frac{dW}{dt}$, which is the rate of discharge of the nozzle at the given temperature and pressure, and is equal to the weight of gas discharged per second. Substituting this value for W in the equations

$$W = C_1 \frac{P_1}{\sqrt{T_1}}, \qquad (1)$$

or

$$W = C_2 \sqrt{\frac{P_2(P_1 - P_2)}{T_1}}, \qquad (2)$$

and solving for C_1 or C_2 we may obtain the coefficient of the nozzle. Equation (1) is to be used in case the pressure in the reservoir, P_1, is greater than

$$P_2 \left(\frac{\gamma-1}{2}\right)^{\frac{\gamma}{\gamma-1}},$$

where P_2 is the pressure of the atmosphere. Equation (2) is to be used in case the pressure in the reservoir is less than the value given. Several computations may be made from one set of observations and the mean of the results taken as the value of the coefficient C_1 or C_2.

It is not usually wise to depend upon the results given by the theoretical equations for the flow of air through orifices, since, as it passes through an orifice, the air stream undergoes a certain amount of contraction which cannot be measured directly. It is usually better, no matter what the form of orifice, to determine its rate of discharge experimentally for several different pressures and to derive from these rates the coefficient C_1 or C_2 which can be substituted in equations (1) and (2) in the preceding paragraph.

PROBLEMS

1. A mass of gas which initially occupies a volume of 2 cu.ft., and has a pressure of 100 lbs. absolute per square inch, expands isothermally. Find its pressure when its volume becomes 4 cu.ft. When it becomes 6 cu.ft. When it becomes 8 cu.ft.

Ans. 50, $33\frac{1}{3}$, and 25 lbs. per square inch.

2. Find the volume of the above mass of gas when the pressure becomes 75 lbs. absolute. When it becomes 40 lbs. gage. When it becomes 20 lbs. gage.

Ans. 2.67, 3.66, and 5.77 cu.ft.

3. Draw the pressure volume curve of the above mass of gas, making the scale of pressures 1 in. equals 25 lbs. per square inch, and the scale of volumes 1 in. equals 1 cu.ft.

4. Find the work done by the mass of gas in Problem 1 in expanding from a volume of 2 cu.ft. to a volume of 6 cu. ft. To a volume of 8 cu.ft.

Ans. 31,600 and 39,900 ft.-lbs.

5. A lb. of air at a temperature of 60° F. expands isothermally from a volume of 3 cu.ft. to a volume of 12 cu.ft. Find the work of expansion. Find the heat added.

Ans. 38,400 ft.-lbs. and 49.4 B.T.U.

6. A lb. of air expands adiabatically and its temperature falls from 200° to 100° F. What is the loss in intrinsic energy in B.T.U.? What is it in ft.-lbs.?

Ans. 16.867 B.T.U. and 13,100 ft.-lbs.

7. A quantity of air expands adiabatically from a pressure of 100 lbs. to a pressure of 25 lbs. per square inch absolute. Its initial volume is 1 cu.ft. What is the final volume. Ans. 2.68 cu. ft.

8. If the initial temperature of the air in problem 7 were 80° F., what would be its final temperature? Ans. −98°F.

9. A mass of air expands adiabatically from a pressure of 100 lbs. The ratio of expansion is 4. Find the final pressure. Ans. 14.25 lbs.

10. If the initial temperature in Problem 9 were 100° F., find the final temperature.

Ans. −142° F.

11. A quantity of air is compressed adiabatically until its temperature rises from 530° absolute to 1000° absolute. If its initial pressure is 1 atmosphere, find its final pressure in atmospheres. Ans. 9.04 atmospheres.

12. If the initial volume of the air in Problem 11 is 1 cu.ft., find its final volume.

Ans. .209 cu.ft.

13. Find the work done by the expanding air in problem 7.

Ans. 11,670 ft.-lbs.

14. One lb. of air is compressed adiabatically, and its temperature is raised from 60° to 250°. Find the work of compression. Ans. 24,900 ft.-lbs.

15. Find the work done by 1 lb. of air expanding as in Problem 10.

Ans. 31,700 ft.-lbs.

16. Find the work required to compress two lbs. of air between the temperature limits given in Problem 11. Ans. 462,000 ft.-lbs.

17. Compute the initial volume of the air in Problem 14, assuming that the initial pressure is 15 lbs. per square inch, and draw the pressure volume curve making the scale of pressure 1 in. equals 15 lbs. and the scale of volumes 1 in. equals 2 cu.ft.

18. One lb. of air having an initial volume of 6 cu.ft. and a pressure of 30 lbs. per square inch absolute expands at constant pressure until its volume is 12 cu.ft. Compute the initial and final temperatures. Ans. ~~16° F. and 502° F.~~

19. Compute the work of expansion in Problem 18. Ans. 25,950 ft.-lbs.

20. Compute the quantity of heat added to the gas in Problem 18.

Ans. 115.2 B.T.U.

21. Compute the increase in intrinsic energy in Problem 18. Ans. 81.9 B.T.U.

22. A quantity of air expands and the work of expansion is twice the loss in intrinsic energy. Find the value of the exponent N. Ans. 1.2034.

23. Computation from the card given by an air compressor shows that the index of the compression line is 1.35. Find the ratio of the work performed to the gain in intrinsic energy. Ans. 1.162.

24. Air is compressed from a pressure of 1 atmosphere to a pressure of 4 atmospheres. The value of the index of polytropic compression is 1.25. Find the final volume in per cent of the initial volume. Ans. 33%.

25. If the initial temperature in Problem 24 be 70° F, find the final temperature.

Ans. 240° F.

26. One pound of hydrogen expands from a volume of 3 cu.ft. to a volume of 10 cu.ft., the index of polytropic expansion being 1.1. Find the final absolute pressure in per cent of the initial absolute pressure. Ans. 26.6%.

27. If the initial temperature is 1000° absolute in Problem 24, find the final absolute temperature. Ans. 886° abs.

28. The initial pressure and volume of 1 lb. of air are 10 atmospheres and 2 cu.ft.

respectively. Compute the pressure when the temperature has fallen to 60° F. as a result of polytropic expansion, the index being 1.3. Ans. 1.555 atmospheres.

29. Compute the volume of the air in Problem 28 when the temperature has fallen to zero F. Ans. 12.50 cu.ft.

30. Draw the expansion line of the air in Problem 28 taking for the scale of volumes 1 in. = 2 cu.ft. and for the scale of pressures 1 in. = 2 atmospheres.

31. Compute the velocity of sound in hydrogen when the temperature of the gas is zero F. Ans. 4010 ft. per sec.

32. The velocity of sound in air is 1125 ft. per second at a temperature of 60° F. Compute the value of γ.

33. An explosive mixture of gas is confined at a pressure of 100 lbs. per square inch. After ignition, the pressure rises to 300 lbs. per square inch. The temperature of the gas is 400° F. Find the rate of propagation of the explosion wave. Take γ as 1.40, and R as 53.0. Ans. 2330 ft. per sec.

34. The minimum area of a nozzle is 1 sq.in. and the velocity of the air passing this section is 2000 ft. per second. Find the volume passing this section per second. Ans. 13.9 cu.ft. per sec.

35. The temperature of the air entering a nozzle is 100° F. Find the temperature of the air in the throat of the nozzle. Ans. 5° F.

36. The pressure of the air as it enters this nozzle is 100 lbs. per square inch absolute. Find the pressure in the throat of the nozzle. Ans. 52.5 lbs. per square inch.

37. Find the velocity of the air passing the throat of the nozzle in Problem 36. Ans. 1064 ft. per second.

38. What will be the highest pressure of the air in the region into which the nozzle discharges, which will permit answers to Problems 35–37 to be correct? Ans. 52.5 lbs.

39. The area of the throat in Problem 36 is 0.001 sq.ft. Find the weight of air discharged per second. Ans. 0.325 lbs.

40. Find the value of k_1 in equation (24), Art. 57, for carbon dioxide. Ans. 0.640.

41. Find the value of the constant C_1 in equation (26), Art. 57, for a nozzle having an area of 0.01 sq.in for carbon dioxide. Ans. 0.0064 when P_1 is in pounds per square inch absolute.

42. Air flows through a nozzle having an area of 0.01 sq.in. from a region where the pressure is 100 lbs. per square inch into a region where the pressure is 90 lbs. per square inch. The initial temperature is 80° F. Find the theoretical rate of discharge. Ans. 0.0164 lbs. per second.

43. Compute the value of k in equation (9) Art. 59, for the conditions in Problem 42. Ans. 1.27.

CHAPTER IV

THERMODYNAMIC PROCESSES AND CYCLES

61. General Definitions. The **thermodynamic state** of a substance is defined when its temperature, pressure, density, mass and composition are known.

A substance having a definite mass, and of homogeneous composition, is, in thermodynamics termed a **body,** no matter what the physical state of the substance may be.

A body is in **thermal equilibrium** when every part of it is of the same temperature.

A body is in **thermodynamic equilibrium** when every part of it is of the same temperature, pressure, and density.

An **isolated body** is one which is so situated that it neither gains nor loses heat.

An **isolated** system, usually termed simply a **system,** is a group of bodies so situated that the system neither gains nor loses heat.

When a body undergoes a change in thermodynamic state, it is said to undergo a **process.** If, at every instant during this process all parts of the body are in thermodynamic equilibrium, the process is said to be **reversible.** If at any point during the process all parts of the substance are not of the same temperature, pressure, and density, the process is said to be **irreversible** or **sweeping.** A substance which undergoes a reversible process can be brought back to its initial state by causing it to pass in the reverse order through each successive state of the process. This is called reversing the process. A substance which undergoes a sweeping process cannot be brought back to its initial state by reversing the sweeping process.

62. Examples of Processes. As an example of a reversible process, we may take the adiabatic expansion of a gas, a process which lowers its temperature and pressure and increases its volume. At every instant during the expansion every portion of the gas has the same temperature, pressure, and density. By raising the pressure upon the gas, its temperature and density will be increased, and its volume decreased, and it will be brought back to its initial state, after passing successively through each state through which it passes during its expansion.

As an example of a sweeping process, we may take the case of a body

of gas confined under pressure, which is allowed to escape from the containing vessel through an orifice. While it is escaping, that portion of the gas which has passed through the orifice will be of a different temperature, pressure, and density from that portion yet within the containing vessel, and the gas is not in thermodynamic equilibrium during the process. Neither is there any possible method which will enable us to pass the gas from a region of low pressure back through the orifice into a region of high pressure, and yet have it pass in turn through the several conditions through which it passed during the progress of the sweeping process, and the process cannot be reversed.

63. Cycles. When a substance is caused to undergo a series of processes, for any purpose, and is finally brought back to the thermodynamic state which it had initially, the substance is said to perform a **cycle.** The substance performing the cycle is called the **working substance,** and in case this substance is a fluid, as it usually is, it is called the **working fluid.** In case the series of processes which the working fluid undergoes in any cycle are all reversible processes, the cycle is said to be **reversible,** or **perfect.** In case one or more of the processes are sweeping processes, the cycle is said to be **irreversible,** or **imperfect.** The purpose of causing a working substance to perform a cycle is usually either to transform heat into work, or to transfer heat from a region of low temperature to a region of high temperature. A thermodynamic machine in which a cycle is performed for the purpose of transforming heat into work is called a **heat engine.** A machine in which a cycle is performed for the purpose of transferring heat from a region of low temperature to a region of high temperature is called a **refrigerating machine.**

64. Entropy. It is convenient in the development of proofs for many thermodynamic theorems, and in working out certain classes of problems, to make use of a ratio or abstract quantity, to which the term **entropy** has been applied.

The entropy of a substance having a given state may be defined as the sum of the quotients obtained by dividing the successive increments of heat necessary to bring the substance to the given state, by the absolute temperature at which each addition of heat occurred, the substance being in a state of thermodynamic equilibrium throughout the several processes necessary to bring the substance to its final state.

This definition may be represented by the equation

$$N = \Sigma \frac{\Delta H}{T},$$

where N is the entropy of the substance, ΔH represents an elementary quantity of heat added to the substance, and T is the absolute temperature of the substance during such addition of heat.

Were it possible to compute the quantity of heat necessary to bring a substance from a temperature of absolute zero to any given state, and to know the absolute temperature at which each successive addition of heat occurred, it would be possible to compute the absolute entropy of the substance.[1] Since, however, this is neither possible nor even desirable, it is customary to compute the change of entropy of a substance in passing from some standard thermodynamic state to any given state. The standard state is then termed the state of zero entropy, or, for brevity, the zero state, and the change of entropy is termed the entropy of the substance.

It may be that in causing a substance to pass from the zero state to some other state, heat must be abstracted instead of being added. In such a case, the entropy change is negative and the entropy of the substance will be a negative quantity. The zero state is usually so chosen that in any series of changes which the substance must undergo while it is the subject of investigation, the entropy will remain a positive quantity throughout this series of changes.

65. Dimensions of Entropy. In dealing with entropy, the student must bear in mind that it is an abstract quantity; a mere ratio, having no physical existence. It is of the same dimensions as the angular functions, or any similar mathematical quantity. It is found by dividing heat, which is energy, and whose dimensions are therefore force $\times$ distance, by temperature, which is shown by the kinetic theory of gases to be energy per molecule (see Chapter XXVI.), and whose dimensions are also therefore force $\times$ distance. It follows therefore that entropy is without physical dimensions, or, as we say, it is an abstract quantity.

66. Proposition I. The entropy of a system is equal to the sum of the entropies of the several bodies composing the system. Let the system consist of several bodies and also let it have the zero state when each of the bodies composing it has the zero state. To bring the system from the zero state to a given state, it is necessary to bring each of the bodies composing it from the zero state to a given state. In doing so, it is necessary to make to each body certain heat additions at certain temperatures, thereby increasing the entropy of the body by a definite amount. Since these heat additions are also made to the system at these same temperatures, the entropy of the system has been increased by the same amount. Obviously, then, the amount by which the entropy of the system will be increased will be the sum of the amounts by which the entropies of the several bodies are increased, or the entropy of the system is the sum of the entropies of the several bodies composing it.

67. Proposition II. The entropy of an isolated system which remains continuously in thermodynamic equilibrium is a constant quantity. This follows from the fact that in order to change the entropy of a body or of

[1] The absolute entropy of a body is always infinity.

a system, heat must be added to or abstracted from the body or system. Since heat cannot be added to or abstracted from an isolated system, its entropy will remain unchanged, provided that it remains continuously in thermodynamic equilibrium.

68. Proposition III. The entropy of a body is unchanged by causing it to complete a reversible cycle. This will be apparent from the following considerations. Assume an isolated system composed of several bodies, each one having a definite entropy. Let us assume that one of these bodies is caused to complete a reversible cycle, during which it, and all other bodies of the system, remain continuously in thermodynamic equilibrium. At the completion of this cycle, assume that the body is taken from the system and replaced by a body identical with it in every respect (i.e., having the same thermodynamic state) but having the same entropy as the body which was taken away originally had. In taking this body from the system and replacing it with an identical one, heat is neither added to nor abstracted from the system, and the entropy of the system remains unchanged. Therefore the entropy of the body taken from the system must be the same as the entropy of the body introduced into the system. Hence, after undergoing any series of reversible processes and returning to its original state, the entropy of a body will be unchanged.

69. Proposition IV. The entropy of a body is independent of the number and kind of reversible processes to which it may have been subjected in bringing it to a given state. Assume a body having the state indicated by A in Fig. 13. Assume that it passes by route c to state B, route c being composed exclusively of reversible processes. Assume that it is brought back to state A by route d, also composed exclusively of reversible processes. The entropy of the body is the same at the end as at the beginning of the cycle. The entropy of the body at state B is equal to the entropy at the beginning plus the entropy gained during the series of processes c. The entropy of the body at the end of the cycle is equal to the entropy at state B minus the entropy lost during the series of processes d. It therefore follows that the entropy which would be gained in passing from state A to state B by route d, is equal to the entropy gained in passing by route c, since the entropy gained in passing from A to B by route d is equal to the entropy lost in passing from B to A by the same

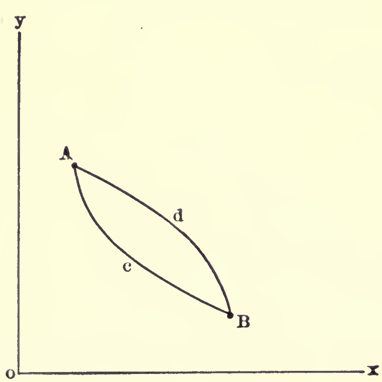


FIG. 13.

route, and this in turn is equal to the entropy gained in passing from A to B by route c. As the only conditions imposed were that each route should be a series of reversible processes, it follows that the entropy gained in passing from A to B will be the same, whatever the series of reversible processes employed.

70. Proposition V. The entropy of a body depends upon its state and it therefore is independent of the processes, reversible or otherwise, by means of which it has been brought to that state. This may be demonstrated as follows: Assume a system of bodies all of which are in the zero state, and assume that one body is caused to pass by any series of processes, reversible or otherwise, to any given state, A. Assume that a body, identical in mass and composition with the one considered, is brought from the zero state to state A by a series of reversible processes. Its entropy will then be a definite quantity. If this second body be substituted for the first, the entropy of the system will remain unchanged, and therefore the entropy of the body withdrawn is the same as the entropy of the body substituted. Hence the entropy of the body withdrawn depends upon its state, and not upon its previous history.

71. The Entropy of a System is Increased by a Heat Transfer. When two bodies of different temperatures composing an isolated system are so placed that heat may pass from one to the other, it will, by definition (see Art. 15) pass from the hotter to the colder body. The amount of heat gained by the latter is equal to the amount of heat lost by the former. Let us call the temperature of the hot body T_1, and that of the cold body T_2, and the quantity of heat transferred H. Then as a result of the heat transfer, the entropy of the hot body will be diminished by the amount

$$\frac{H}{T_1},$$

and the entropy of the colder body will be increased by the amount

$$\frac{H}{T_2}.$$

Since the latter quantity is greater than the former (its denominator being less than the denominator of the former quantity, while the numerators are equal), the entropy of the system has been increased. We may therefore state that when a heat exchange occurs between bodies of different temperatures composing a system, the entropy of the system is thereby increased.

72. The Entropy of a System is Increased by a Sweeping Process. When a body of a system undergoes a sweeping process, the entropy

of the system is thereby increased. A body which undergoes a sweeping process must suffer, as a result, a change in temperature and pressure, in temperature and volume, in pressure and volume, or in all three. If the body undergoes a change in temperature and pressure, its volume remaining unchanged, it must either have heat added to it or have heat taken away from it. In either case, heat must be transferred from a region of high temperature to a region of low temperature, and the process will result in an increase in the entropy of the system.

In case the body suffers a change in volume and pressure as a result of a sweeping process, the final volume must, of necessity, be greater than the initial volume. In order to return the body to its initial state, work must be done upon it and heat extracted from it. The amount of heat so extracted divided by the absolute temperature of the body measures the increase in entropy of the system resulting from the sweeping process.

In case the temperature and volume of the body are increased, the pressure remaining constant, heat must be added to the body by some body having a higher temperature with resulting increase in the entropy of the system. In case the temperature and volume are diminished heat must be abstracted from the body by some body having a lower temperature with resulting increase in the entropy of the system.

In case a body undergoes a simultaneous change of pressure, temperature and volume, the entropy must be increased, since this change will be the equivalent of two changes, either one of which will increase the entropy of the system.

It therefore follows that when a sweeping change occurs within a system, the entropy of the system is thereby increased.

73. The Entropy of a System Cannot be Diminished. It follows from the facts which we have been developing in regard to entropy, that the entropy of a system can never be diminished, but must continually increase as the bodies of the system undergo sweeping processes. This is another way of stating the well-known truth that all forms of energy tend to become heat and that heat tends to distribute itself until all bodies have the same temperature. A recognition of these facts is so fundamental to the science of thermodynamics that some form of statement of them is usually called the second law of thermodynamics, the so-called first law of thermodynamics being a statement of the interrelation of heat and other forms of energy.

74. The Carnot Cycle. The simplest form of reversible cycle is that devised by the French engineer Carnot, and known therefore as the Carnot cycle. In the Carnot cycle, the working substance undergoes the four reversible processes indicated in the diagram in Fig. 14. First, the substance is caused to expand isothermally at some temperature T_1, from

state 1 to state 2. It is then caused to expand adiabatically from state 2 to state 3, thereby attaining some lower temperature T_2. It is then compressed isothermally to state 3 and finally compressed adiabatically until it reaches its initial state. Since the several reversible processes may be passed through in reverse order, the cycle itself is a reversible cycle.

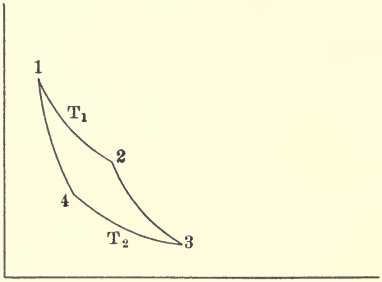


FIG. 14.

75. The Carnot Engine. In order to understand Carnot's cycle, it will be well to devise an imaginary apparatus capable of working upon this cycle. Such an apparatus, illustrated in Fig. 15, will consist of a cylinder of some non-conducting material in which moves a piston operating a slider-crank mechanism. The head of the cylinder is made of some material which is a good conductor of heat. The space between the piston and cylinder head is occupied by the working substance, which will have at the beginning of the cycle a temperature T_1. A large hot body of the same temperature, which acts as a source of heat, and

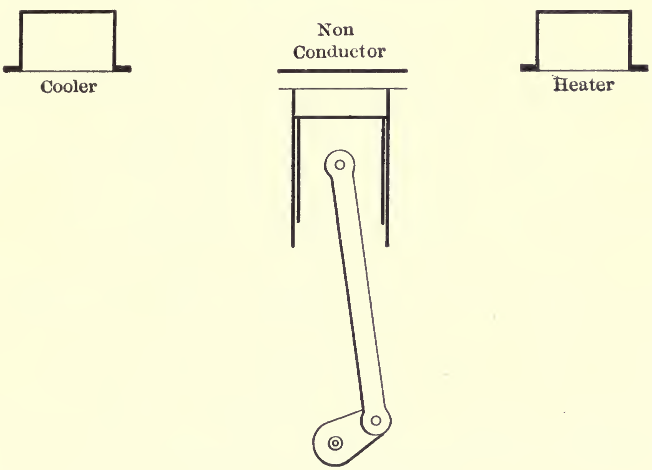


FIG. 15.—Carnot's engine.

which is called the heater, is applied to the conducting head of the cylinder, and the working substance allowed to expand. The temperature is thereby lowered an infinitesimal amount, and heat will immediately flow from the heater into the working substance, in order to maintain the temperature at the value T_1. After the expansion has proceeded sufficiently, the heater is removed and replaced by a non-conductor of heat. As the

working substance continues to expand, and as no heat is supplied, the temperature now begins to fall. When this adiabatic expansion has proceeded until the temperature reaches the value T_2, a body having the temperature T_2, and which acts as a cooler, is applied to the conducting head of the apparatus, and the substance is compressed. As a result, its temperature will be raised by an infinitesimal amount, and heat will flow from the working substance into the cooler, maintaining the temperature of the working substance constant, at the value T_2. When the isothermal compression has proceeded sufficiently, the cooler is replaced by the non-conducting substance and the working substance is compressed adiabatically to its initial state.

76. Efficiency of the Carnot Cycle. During the isothermal expansion, a quantity of heat H_1, whose amount depends upon the mass of the working substance and the amount of expansion which it undergoes, is imparted to the working substance. During isothermal compression, a quantity of heat H_2 is abstracted from the working substance. The amount of work performed by the substance upon the piston of the engine is obviously equal to the mechanical equivalent of the difference between the heat supplied and the heat abstracted. The efficiency of the apparatus is therefore given by the expression,

$$E = \frac{H_1 - H_2}{H_1}.$$

During this cycle the entropy of the system remains constant, since the cycle is composed exclusively of reversible processes and the system remains continuously in thermodynamic equilibrium. Since the working substance completes a perfect cycle, its entropy at the end of the cycle is the same as at the beginning. Hence the entropy lost by the heater must be equal to the entropy gained by the cooler. The entropy lost by the heater is $\frac{H_1}{T_1}$. The entropy gained by the cooler is $\frac{H_2}{T_2}$. Writing them equal, we have

$$\frac{H_1}{T_1} = \frac{H_2}{T_2}.$$

Hence, we will have for the efficiency of the cycle

$$E = \frac{H_1 - H_2}{H_1} = \frac{T_1 - T_2}{T_1}.$$

It will now be shown that no engine can be more efficient than a Carnot engine acting within the same temperature range.

77. The Efficiencies of Other Cycles. If a Carnot engine be driven backward, it will act as a refrigerating machine, taking heat from the cooler, and adding to the heater a quantity of heat equal to that taken

from the cooler plus the heat equivalent of the work supplied. Assume an engine operating by a cycle having a greater efficiency than a Carnot cycle. If this engine does the same work, it will be able to drive the Carnot engine backward. The Carnot engine will then extract from the cooler more heat than is given to it by the engine assumed, and will give off to the heater more heat than is taken from it by the engine assumed. The amount of heat lost by the cooler at the low temperature T_2 will be equal to the amount of heat gained by the heater at the high temperature T_1, and the entropy of the system will be diminished. Since this is impossible, it follows that no cycle can be more efficient than Carnot's cycle.

In like manner, it may be shown that no reversible cycle can be less efficient than Carnot's cycle, for then Carnot's cycle, doing the same work, can drive this cycle backward, transfer heat from the cooler to the heater, and so decrease the entropy of the system, which is impossible.

No imperfect cycle can be as efficient as Carnot's cycle, since the result of a sweeping process is to increase the entropy of the system in which it occurs. At the end of such a cycle, the entropy of the working substance is the same as at the beginning of the cycle, since the working substance has the same thermodynamic state. The entropy of the heater will, of course, be diminished, and the entropy of the cooler increased. Since the entropy of the system is made greater by the cycle, the gain in entropy of the cooler is greater than the loss in entropy of the heater. The loss in entropy of the heater will be

$$\frac{H_1}{T_1}.$$

The gain in entropy of the cooler will be

$$\frac{H_2}{T_2}.$$

The efficiency of the cycle will be

$$\frac{H_1 - H_2}{H_1},$$

and since $\frac{H_2}{T_2}$ is greater than $\frac{H_1}{T_1}$ it must follow that the efficiency of this cycle is less than

$$\frac{T_1 - T_2}{T_1}.$$

We may therefore conclude the following in regard to the efficiency of heat engines: First, all heat engines operating on perfect cycles have the same efficiency for the same temperature conditions. Second, the

higher the temperature at which the engine receives heat, and the lower the temperature at which it rejects heat, the greater the efficiency of the engine. Third, no engine operating on an imperfect cycle can be as efficient as an engine operating on a perfect cycle. Fourth, it will be noted that in our discussion of the Carnot cycle and of other cycles, no assumptions were made in regard to the nature of the working substance, hence the efficiency of an engine operating by a perfect cycle is independent of the nature of the working substance, and depends only on the temperature limits between which the engine operates. From the equation giving the efficiency of a reversible cycle, it will be seen that the only method by which we may increase the efficiency of such a cycle is by increasing the temperature at which it receives its heat, or by decreasing the temperature at which it rejects its heat. Fifth, the efficiency of an imperfect cycle may depend on the nature of the working substance.

78. The Efficiency of the Thermo-couple. It is known that if two unlike metal rods be joined at two points and one junction heated while the other junction is cooled, an electromotive force is produced, and a current is caused to pass through the rods. Such a device is known as a thermo-couple. The passage of the current through the cold junction tends to heat it while the passage of the current through the hot junction tends to cool it. Heat must therefore be supplied to the hot junction to maintain its temperature, and abstracted from the cold junction for the same reason. The quantity of electrical energy generated is equal to the quantity of heat supplied to the hot junction less the quantity of heat rejected by the cold junction. The thermodynamic efficiency of the apparatus is the same as that of any reversible cycle, namely,

$$E = \frac{T_1 - T_2}{T_1}$$

where T_1 is the temperature of the hot junction and T_2 is the temperature of the cold junction. Were this not so, such an apparatus could be made to drive a Carnot engine backward or be operated as a refrigerating machine by a Carnot engine, according as it is more or less efficient than the Carnot engine, and so transfer heat from the cooler to the heater. Such an action is impossible, since it would decrease the entropy of the system. It will thus be seen that the thermo-electric couple obeys the same laws as does any thermodynamic engine, and, by a similar process of reasoning, it may be shown that any method of transforming heat into work must also obey these laws.

79. The Vital Processes not Thermodynamic. In the bodies of animals, the potential chemical energy of food is transformed into mechanical work by some method at present unknown to us. Since the temperature of all parts of an animal body is constantly maintained at or about 98° F., and there can be no appreciable difference of temperature in the different parts, the energy of the food is not transformed into heat before being transformed into mechanical energy, but is transformed into work by some process which is not thermodynamic in its nature. Experiment shows us that the efficiency of an animal as a machine for the transformation of potential chemical energy into mechanical energy ranges from 30 to 50 per cent, which is a much greater efficiency than it is possible to realize by any machine man has as

yet constructed. Were it possible to discover the method by which chemical energy is transformed in the animal body, we might build machines of much greater efficiency than those we now possess, but obviously they would not be thermodynamic machines.

80. The Regenerative Principle. In certain kinds of engineering apparatus, whose various parts taken together form a system, it is often useful to make use of a principle termed regeneration. Regeneration is the storage of heat in some conducting body, the heat being taken from some substance which is being rejected from the system, and subsequently imparted to some substance which is being introduced into the system. As an example of the use of the regenerator, we may take the method employed in steel works in saving the heat which would be otherwise lost from open-hearth furnaces. In Fig. 16, *A* is the combustion chamber of such a furnace. In this combustion chamber highly heated gas and air are brought together, and the resulting combustion still

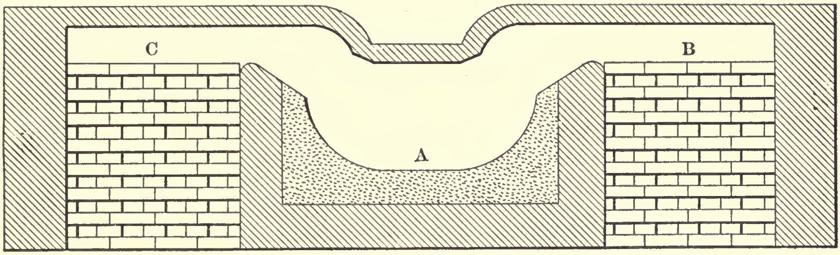


FIG. 16.—Regenerative Furnace.

further increases their temperature. The current of gas and air being from the left to right, it will leave the combustion chamber and pass through the chamber *B*, which is filled with brick checker work (i.e., brick work built up in such a way as to afford passage for the gases). In their passage through this checker work the gases heat the brick, that part of the checker work near the furnace chamber being brought almost to the temperature of the furnace and that part near the breeching into the which gases pass, being warmed somewhat. Were the gases allowed to pass for a considerable length of time in this direction, the whole of the checker work would eventually reach the temperature of the furnace, but they are not allowed to pass through for any considerable period. After a few minutes, the direction of the current of gases through the apparatus is reversed, gas and air passing through the checker work from right to left, into the combustion chamber, where they are burned and then pass out, heating the checker work in chamber *C*. The gases which enter the checker work in chamber *B* have their temperature slightly raised by the first bricks which they encounter. As they progress from right to left they continue to receive heat, since they encounter hotter and hotter brick

in their progress, and at length they enter the combustion chamber at almost the temperature of the chamber. By this means we are enabled to maintain a very high temperature in the combustion chamber without rejecting gas of a high temperature, which would entail very great heat losses.

If an engine be caused to work by a cycle in which the working substance is transferred from a region of high temperature to a region of low temperature, passing through a regenerator on the way, the cycle is said to be a regenerator cycle. Such cycles may be made to have very high efficiencies, approaching, in fact, the efficiency of the Carnot cycle. An example of such a cycle is given in the description of the Stirling hot-air engine in Chapter XVIII.

81. Pseudo-Cycles. In most practical thermodynamic machines the working substance is rejected from the cylinder or working chamber during some portion of the cycle, and is replaced by a fresh portion of working substance during some other portion of the cycle. Since the working substance is not returned to its original state, it will be apparent that it does not undergo a true cycle, but merely a series of processes. Such a series of processes may be termed a pseudo-cycle. It may be that the Watt diagram of such a series of processes can, in theory, be reproduced exactly by causing a definite mass of working substance to undergo a cycle while it remains continuously within the working chamber. If such is the case, the pseudo-cycle is the exact equivalent of a true cycle and may, in the thermodynamic computation, be treated as such. The Otto gas engine cycle (see Chapter XIX.) is a cycle of this type. On the other hand, however, the series of processes is often of such a nature that its Watt diagram cannot be reproduced within a working chamber containing a constant mass of working fluid, and it is not equivalent to a true cycle. Most practical cycles are of this type, and an important class of thermodynamic machines, usually termed gas compressors, invariably operate upon such pseudo-cycles. They are employed for the purpose of transferring fluid from a region of low pressure to a region of high pressure, and the rejection of the working fluid during some portion of the cycle, and its replacement by a fresh charge, is an essential process of the machine.

An example of a perfect cycle has already been given in the case of the Carnot cycle. The Otto gas engine cycle, described in Chapter XIX., is an illustration of an imperfect cycle. The Stirling hot-air engine described in Chapter XVIII., is an example of the use of the regenerator cycle. The ordinary air compressor described in Chapter XXII. is an example of a pseudo-cycle.

PROBLEMS

1. Twenty B.T.U. are imparted to a substance whose absolute temperature is 1000°. How much is its entropy increased? Ans. 0.02.

2. 10.592 B.T.U. are imparted to a substance whose temperature is 70° F. By what amount is its entropy increased? Ans. 0.02.

3. The specific heat of a substance is unity. One pound of it is raised in temperature from 500° absolute to 600° absolute. What is its increase in entropy? Ans. 0.1823.

4. The specific heat of a substance is 0.2 and its mass is 10 lbs. By how much was its entropy increased in raising it from 60° to 80° F.? Ans. 0.0784.

5. Two pounds of water, 1 lb. of which has a temperature of 500° absolute and the other of which has a temperature of 600° absolute are mixed together. How much is the entropy of the system increased? (Assume the specific heat of water to be unity.) Ans. 0.0083.

6. One pound of air is confined in a volume of 1 cu.ft. at a temperature of 500° absolute. It is allowed to expand suddenly to a volume of 3 cu.ft. without doing work. What quantity of work must be done upon it to return it by isothermal compression to its initial state? Ans. 29,300 B.T.U.

7. How many B.T.U. must be taken from it during this process?

Ans. 37.69 B.T.U.

8. What was the increase in entropy resulting from the sudden expansion?

Ans. 0.07538.

9. A Carnot cycle engine takes heat at 300° F. and rejects it at 80° F. What is its efficiency? Ans. 29%.

10. The working fluid in the above engine is 1 lb. of air whose initial volume is 2 cu.ft. Find its volume at the beginning of adiabatic compression.

Ans. 4.66 cu.ft.

11. The ratio of isothermal expansion in this cycle is 2. Find the volume of the air at the end of adiabatic expansion. Ans. 9.32 cu.ft.

12. What quantity of heat was imparted to the air during isothermal expansion?

Ans. 36.1 B.T.U.

13. What quantity of heat was rejected by the air during isothermal compression? Ans. 25.7 = B.T.U.

14. What quantity of work was done by the engine during the cycle?

Ans. 8030 ft.-lbs.

(Note the relation between the heat supplied and the heat rejected and the work done and also between the heat supplied, the work done and the efficiency.)

15. The hot junction of a thermo-couple is maintained at a temperature of 400° F. and the cool junction at a temperature of 32° F. What is the efficiency of the couple?

Ans. 32.8%.

16. The average temperature of the waste gases entering a regenerator is 1200° F. The average temperature of the air drawn from the regenerator is 1100° F. The temperature of the air entering the regenerator is 60° F. Find the efficiency of the regenerator. Ans. 91.2%.

CHAPTER V

THE THERMAL PROPERTIES OF VAPORS

82. The Effects of the Addition of Heat to Water. A vapor has been defined in Art. 23, as an elastic fluid which may be readily condensed into a liquid by a slight reduction in temperature. We may best understand the constitution of vapors if we study the manner of their formation, and for this study we will take steam as a typical vapor. Assume that we have confined in the apparatus illustrated in Fig. 17, which is identical with the apparatus previously described in Art. 24, one pound of pure water at a temperature of 32° F. This water will be found to occupy a volume of 0.01603 cubic feet. We will assume that there is placed upon the piston a weight of 2116.3 pounds, which, of course, produces a pressure of one atmosphere. If heat be applied to the apparatus it will be found that the water increases in temperature and expands slightly in volume. When 180 B.T.U. have been added to the water, it will, of course, have a temperature of 212° F., and its volume will be 0.0168 cubic feet.

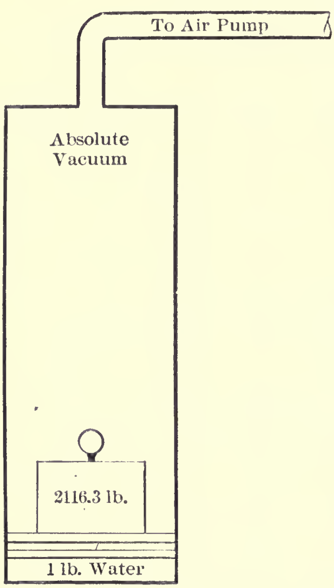


FIG. 17.—Ideal apparatus for the investigation of the properties of vapors.

If we then continue to add heat to the water we will find that its temperature no longer rises, but that some of it is changed into an elastic fluid, which we term steam. The temperature of the steam is exactly the same as the temperature of the water from which it is formed, namely, 212° F. As we continue to add heat, more and more of the water is evaporated, until finally when the total amount of heat added from the beginning of the experiment is 1150.4 B.T.U., the last bit of water is evaporated. The volume of the steam formed at this pressure from the pound of water is found to be 26.79 cubic feet.

From the time that the water reached the temperature of 212° until the last bit of it evaporated, the temperature remained constant, in spite of the fact that 970.4 B.T.U. were added to it during this period. A part of the energy so added was of course expended in lifting the weight upon the piston a distance of 26.77 feet. The remainder of the energy was expended in separating the molecules of water from each other against the forces exerted by their mutual attractions. The first quantity, which is 56,700 foot pounds, or 72.8 B.T.U., is known as the external work of evaporation, while the second quantity, which is the difference, or 879.6 B.T.U., is known as the internal energy of evaporation. Since the whole of this heat added during the evaporation of the steam produces no rise in temperature, but is transformed into some other form of energy, which is in the nature of potential energy, the heat so transformed is termed latent heat, and the sum of the internal energy and the external work is known as the latent heat of evaporation.

83. The steam contained in the cylinder, when free from water, is said to be dry. Since it is at the same temperature as the water from which it was formed, and the slightest reduction in temperature would recondense it into water, it is also said to be saturated. If this dry and saturated steam be heated still further, we will find that its temperature begins to rise, and that it takes approximately 0.47 B.T.U. per pound of steam to raise its temperature one degree. As more and more heat is added, it will be found that the temperature of the steam continues to rise, but that the rise in temperature is not strictly proportional to the amount of heat added. Careful experiments indicate that to raise its temperature from 213° to 313° requires 46.9 B.T.U., to raise it from 313° to 413° requires 46.8 B.T.U., to raise it from 413° to 513° requires 46.7 B.T.U.

84. Vaporization at Other Pressures. Had we heated one pound of water in this apparatus under a different pressure, say for instance 14,400 pounds per square foot, the phenomena observed would have been of exactly the same kind.

The water would be heated to a temperature of 327.8° before steam began to form; 298.3 B.T.U. would have been necessary to produce this rise in temperature. After the addition of 888.0 more B.T.U., the last bit of water would evaporate and the steam formed would be found to occupy a volume of 3.012 cubic feet. Any further addition of heat would then increase the temperature of the steam. To raise the temperature from 327.8° to 427.8° would require 54.5 B.T.U. To raise it from 427.8° to 527.8° would require 59.7 B.T.U. To raise it from 527.8° to 627.8° would require 58.4 B.T.U.

85. Temperature of Vaporization. In general, from such an experiment, or from other experiments designed to investigate the same phenomena, it may be shown that the phenomena produced by the addi-

tion of heat to one pound of water confined under a constant pressure are invariably as follows: When water is confined under any given pressure, the addition of heat increases its temperature, and it boils when the temperature reaches a definite value. This temperature is known as the **temperature of vaporization** of steam of the given pressure, since the steam formed has the same temperature as the water from which it is formed. The symbol used for the temperature of vaporization is engineering work, (for instance in steam tables and thermodynamic equations) is t, and it is expressed, in engineering work, in degrees F. The symbol used in engineering work for the pressure of steam is p, and the pressure is expressed in pounds per square inch absolute.

86. Heat of the Liquid. The amount of heat required to raise one pound of water from the ice point to any given temperature of vaporization,[1] without the formation of steam, is a definite quantity. This quantity of heat is known as the **heat of the liquid.** The symbol used for the heat of the liquid in engineering work is h,[2] and it is expressed in B.T.U.

87. Vaporization. Upon the further addition of heat to water having the temperature of vaporization corresponding to the pressure under which it is confined, it is transformed into steam without further increase in temperature, so long as any water remains untransformed. After all the water is transformed into steam, any further addition of heat increases the temperature of the steam.

Steam which has the same temperature as that of the water from which it was formed (i. e. which has the temperature of vaporization corresponding to its pressure) is said to be **saturated,** since any reduction in temperature will condense some of it into water, and reduce the pressure. Steam which does not contain any water suspended in it in the form of drops or mist is said to be **dry.** Steam which contains no suspended moisture, and yet has the temperature of vaporization corresponding to its pressure, is said to be **dry and saturated.**

88. Specific Volume and Density. The volume of 1 pound of dry and saturated steam at any given pressure is a definite quantity. This volume is known as the **specific volume of steam** of that pressure (or corresponding temperature). The symbol for specific volume is V, and the specific volume is expressed in engineering work in cubic feet.

The weight of one cubic foot of dry and saturated steam of any pres-

[1] The exact amount of heat required varies slightly with the pressure conditions. The water may be heated under the pressure at which it is finally vaporized, as in the ideal experiment described in Art. 82. It may be heated under the pressure produced by its own vapor, which grows steadily greater as the temperature rises. Or it may be heated under barometric pressure, until its temperature corresponds to this pressure, and thereafter be heated under the pressure of its own vapor. The last method is the one usually assumed to be employed.

[2] In some cases the symbol q is used.

sure is termed the **density of the steam** at that pressure, (or corresponding temperature). The density of steam is the reciprocal of its specific volume at the same pressure, and is therefore a definite quantity for any given pressure. It is indicated by the symbol $1/V$, and is expressed in pounds per cubic foot.

89. Latent Heat. The quantity of heat required to vaporize completely 1 pound of water into dry and saturated steam of the same temperature as the water, is a definite quantity for any particular temperature. This quantity of heat is called the **latent heat of evaporation** of steam of the given temperature (or corresponding pressure). The symbol for the latent heat of evaporation is L [1], and it is expressed in engineering work in B.T.U.

90. Total Heat. The quantity of heat required to raise 1 pound of water from the ice point to any temperature of vaporization and to evaporate it into dry and saturated steam having that temperature, is a definite quantity. This quantity of heat is called **the total heat of the steam** at the given temperature (or corresponding pressure), and is expressed in B.T.U. The symbol used in engineering work for the total heat of the steam is H.

91. External Work. The work done by 1 pound of steam in expanding from the volume occupied by 1 pound of water of any temperature to the specific volume of dry and saturated steam of the same temperature, against the pressure corresponding to this temperature, is called the **external work of evaporation** of steam of the given temperature (or corresponding pressure). Since both the change in volume and the pressure are definite quantities for any given temperature, the external work of evaporation is also a definite quantity for any given temperature. The symbol for this quantity is $\frac{144PV}{J}$, and it is expressed in B.T.U.

92. Internal Energy. The difference between the latent heat of evaporation and the external work of evaporation is called the **internal energy of evaporation.**[2] The symbol for the internal energy of evaporation of steam of any given temperature is I and the quantity is expressed in B.T.U. The internal energy of evaporation plus the heat of the liquid is known as the **internal energy of the steam.** The symbol for the internal energy of the steam is E, and this quantity is also expressed in B.T.U.

93. Entropy of Steam. The integral of the quantity $\frac{dh}{T}$ between the limits of 32° F., and any temperature of vaporization is known as the

[1] In some cases the symbol r is used.

[2] The term "disgregational energy" is also used.

entropy of the liquid at that temperature (or corresponding pressure). The symbol for the entropy of the liquid is M.[1] Entropy is an abstract quantity.

The latent heat of evaporation of 1 pound of steam divided by the absolute temperature of the steam is known as the **entropy of evaporation.** The symbol for entropy of evaporation is $\frac{L}{T}$. The sum of the entropies of the liquid and of evaporation is known as the **entropy of the steam** and the symbol for this quantity is N.[2]

94. Steam Tables. All of the quantities enumerated above are of interest and of practical use in engineering work. They are known collectively as the **properties of steam,** and a table giving their values for steam of different pressures or temperatures is known as a table of the properties of steam. Such tables were first accurately computed by the French scientist Regnault, who devoted some years to a study of the best methods of determining these properties, and whose work was of such a high character as to be a model of engineering exactitude. These properties have since been redetermined by many other scientists, who have used in their investigations the most accurate instruments which it was possible to construct. Since in most cases different investigators have used different methods in carrying out these investigations, their work has served as a check upon that of each other and also upon that of Regnault, so that the properties of steam over a considerable range of pressure and temperature are now very accurately determined. These properties have been embodied in two sets of tables available to the English-speaking engineer, one by Marks and Davis, and the other by Peabody. The properties quoted in this book are invariably from the tables of Marks and Davis, unless otherwise stated.

95. Experimental Determination of the Properties of Steam. The properties of dry and saturated steam of a given temperature which are determined by direct experiment, are the pressure, the heat of the liquid, and the total heat of the steam. A steam table usually gives us the properties of steam for each degree of temperature from 32° to 400° F. It must not be inferred, however, that the properties of steam have been determined experimentally for every degree of temperature within this range. Like other physical phenomena, these properties are interrelated by certain natural laws, and therefore their relations may be expressed with great accuracy by empirical equations. A statement of the method by which these properties have been determined experimentally and their values computed for steam tables, will be of interest.

The relation between the pressure and the temperature of saturated steam may be determined experimentally by simultaneously measuring the temperature and the pressure of such steam by suitable instruments. It is necessary, of course, that

[1] The letter ϕ is often used. [2] The letter θ is often used.

these instruments be accurate, that they be properly calibrated, and that, in general, the work be so conducted as to eliminate errors. After determining experimentally a series of values for the pressure of saturated steam of different temperatures, it is necessary to discover an equation expressing the relationship. Many such equations have been proposed, the most accurate of which is that of Marks, which has the following form: [1]

$$\log p = 10.515354 - 4873.71\, T^{-1} - 0.00405096T + 0.000001392964\, T^2$$

where p is the pressure of the steam in pounds per square inch and T is the absolute temperature of the steam in Fahrenheit degrees.

The heat of the liquid may be determined by measuring the quantity of electrical energy used in heating a known weight of water from the ice-point to any required temperature. In conducting such an experiment, it is, of course, necessary to take precautions against many different kinds of errors. No satisfactory formula for the heat of the liquid has yet been produced except the one

$$q = (t-32) + C,$$

in which C is a correction determined from a graphical representation of the results of the experimental work.

The determination of the total heat of the steam is the most difficult part of all the experimental work in this field. The principal difficulty lies in the impossibility of obtaining absolutely dry and saturated steam. However, several methods have been used which give results known to be accurate within $^1/_{10}$ of 1 per cent of the total value of the quantity. The result of these experiments may be represented for temperatures above 212° by the equation,[2]

$$H = 1150.3 + 0.3745(t-212) - 0.000550(t-212)^2.$$

From this equation and from graphical representations of experimental work covering the range below 212°, the value for the total heat of saturated steam may be computed for each degree of temperature within the range which a steam table is intended to cover.

96. The Computation of Properties not Directly Observed. All other properties of steam are determined from the pressure, temperature of vaporization, heat of the liquid, and total heat of the steam by means of the thermodynamic relations of these four quantities. The latent heat of evaporation is obtained by subtracting the heat of the liquid from the total heat of the steam. The entropy of the liquid is found by a step-by-step integration of the quantity $\frac{dh}{T}$, and the entropy of evaporation by dividing the latent heat of evaporation by the absolute temperature of vaporization. The total entropy of the steam is the sum of the entropies of the liquid and of evaporation.

The specific volume is determined in the following manner: Assume that 1 pound of steam is caused to perform a Carnot cycle, between the temperature limits T and $T-dT$. The Watt diagram of this cycle is shown in Fig. 18. At the beginning of this Carnot cycle the cylinder of the Carnot engine will contain 1 pound of water at a temperature T, and under the corresponding pressure P. During the isothermal expansion this water will be entirely evaporated by adding to it the latent heat of evaporation

[1] See the Transactions of the A.S.M.E. for 1911.

[2] See footnote to article 101 for Davis' method of determining the total heat.

at the constant temperature T and the corresponding constant pressure P. When the pound of water is entirely evaporated, it is allowed to expand adiabatically until its temperature falls by the infinitesimal amount dT. It is then isothermally compressed while under the pressure $P-dP$ and at the corresponding temperature $T-dT$. When it is almost entirely condensed, the condensation is stopped, and the remainder of the compression is adiabatic, raising the temperature of the mixture of steam and water to the value T, and condensing the remaining steam. In this process, the quantity of heat imparted to the water is equal to the latent heat of evaporation. The quantity of work done is, of course, VdP, where V is the increase in volume of the steam (i.e., the difference in volume between the pound of dry and saturated steam and the pound of water under the given pressure), and dP is the change in pressure. The efficiency of the cycle is, of course, $\frac{dT}{T}$. Hence we may write

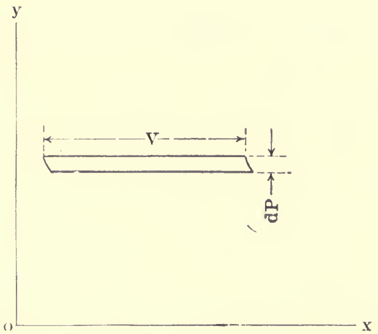


FIG. 18.—Carnot cycle for steam.

$$\frac{L\,dT}{T} = V\,dP \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Solving this for V, we will have

$$V = \frac{L}{T} \times \frac{dT}{dP} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

If we plot from the steam tables a curve showing the relation of the temperature and the pressure of steam (the pressure being in pounds per square foot), we may at any point in this curve draw a tangent, and from the intercepts we may determine the value of the expression $\frac{dT}{dP}$. By means of this value, and the known latent heat of evaporation for the given temperature, we may compute the increase in volume V, and by adding to this the original volume of the water, we obtained the volume of the steam at the given temperature and pressure. After obtaining the specific volume for a number of temperatures, we may construct a curve or derive an equation from which the specific volume of steam of any temperature may be determined. The density of steam is, of course, the reciprocal of the specific volume of the steam.

The external work of evaporation is found by multiplying the change in volume in passing from the condition of a liquid to the condition of dry and saturated steam, by the pressure of the steam in pounds per square foot. This quantity divided is by 777.5 in order to reduce it to B.T.U. The internal energy of evaporation is equal to the latent heat of evaporation minus the external work of evaporation. The internal energy of the steam is equal to the internal energy of evaporation plus the heat of the liquid.

97. The Properties of Other Vapors. The phenomena observed when other liquids than water are evaporated into their vapors are exactly similar to the phenomena observed in the case of water. The quantities

of heat, the pressure, the temperatures, the specific volume, etc., will of course be different for different vapors, but the methods of determining these quantities are the same for all vapors. In the case of such vapors as sulphur-dioxide, ammonia ether, alcohol, chloroform, carbon bisulphide, carbon tetrachloride, and aceton, which are vapors used commercially in refrigerating machines of various types, the properties have been determined with some degree of accuracy and are embodied in tables available to engineers.

PROBLEMS

Find from a steam table the properties of steam asked for in the following problems. The answers are from the tables of Marks and Davis. Other answers will usually be obtained by the use of other tables. Interpolate when necessary.

1. What is the temperature of vaporization of steam at pressures of 1 lb. absolute? 10 lbs. absolute, and 100 lbs. gage? Ans: 101.8°, 193.2°, and 337.9°.

2. What is the pressure of saturated steam at temperatures of 100°, 200°, and 300°? Ans. 0.946, 11.52, and 67.00 lbs. per square inch.

3. Find the heat of the liquid in each case in Problem 1. Ans. 69.8 and 308.8 B.T.U.

4. What quantity of heat is required to raise 1 lb. of water from the ice-point to the several temperatures given in Problem 2? Ans. 67.97, 167.9, and 269.6 B.T.U.

5. What is the volume of 1 lb. of dry and saturated steam at the pressures given in Problem 1? Ans. 339.0, 38.38, and 3.886 cu.ft.

6. What is the density of dry and saturated steam at the temperatures given in Problem 2? Ans. 0.008251, 0.02976, and 0.1547 lbs. per cu.ft.

7. Find the latent heat of evaporation of steam at the pressures given in Problem 1. Ans. 1034.6, 982.0, and 880.0 B.T.U.

8. Find the total heat of steam at the temperatures given in Problem 2. Ans. 1103.6, 1145.8, and 1179.1 B.T.U.

9. Find the internal energy of evaporation of 1 lb. of steam at the three pressures given in Problem 1. Ans. 972.2, 910.9, and 789.1 B.T.U.

10. Find the external work of evaporation at the temperatures given in Problem 2. Ans. 61.5, 71.6, and 79.8 B.T.U.

11. Find the entropy of the liquid at the pressures given in Problem 1. Ans. 0.1327,0.2832, and 0.4875.

12. Find the increase in entropy of 1 lb. of water when it is evaporated into steam at the temperatures given in Problem 2. Ans. 1.8506, 1.4824, and 1.1972.

13. Find the total entropy of 1 lb. of steam at the pressures given in Problem 1. Ans. 1.9754, 1.7874, and 1.5909.

CHAPTER VI

WET AND SUPERHEATED VAPORS

98. Quality of a Vapor. When a vapor contains suspended within it in the form of fine bubbles or drops a quantity of the liquid from which it was formed, the vapor is said to be **wet.** The vapor and the suspended liquid have the same temperature and are under the same pressure, and the whole mass may therefore be said to be in a state of thermodynamic equilibrium, since the division of the particles of liquid is so fine that during expansion or compression the whole mass will not only remain in thermal equilibrium, but it will remain homogeneous in character. The proportion which the dry and saturated vapor present bears by weight to the whole quantity of the mixture, is termed the **quality of the wet vapor** and is usually expressed as a per cent. The symbol for the quality of a wet vapor is q.[1] One pound of wet vapor will therefore consist of q pounds of dry and saturated vapor, and of $1-q$ pounds of liquid. Thus 1 pound of steam of 90 per cent quality contains $^9/_{10}$ of a pound of dry and saturated steam, and there is $^1/_{10}$ of a pound of water suspended in this steam.

If a wet vapor be thermally isolated, its quality will remain constant provided the pressure remains unchanged, but on account of the greater density of the particles of fluid, they will tend to fall to the bottom of the containing vessel, thus separating the wet vapor into two portions, one consisting of dry and saturated vapor, and the other of liquid. This process of course destroys the homogeneity of the wet vapor by separating it into two thermodynamic bodies. Since, however, the diameter of the particles of liquid is exceedingly small, the rate at which they descend through the vapor is also small, and this action goes on but slowly. Consequently, wet vapors when in motion, do not change their quality in any sensible degree during short periods of time.

99. Properties of a Wet Vapor. The heat of the liquid of a wet vapor is the same as the heat of the liquid of the dry and saturated vapor of the same temperature (or pressure).

The latent heat of evaporation of a wet vapor is equal to the latent heat of evaporation of the dry and saturated vapor of the same tem-

[1] The symbol x is often used for this quantity.

perature (or pressure) multiplied by the quality of the wet vapor. This may be expressed by the formula

$$L_w = q\,L,$$

in which L_w is the latent heat of evaporation of the wet vapor, L is the latent heat of evaporation of the dry and saturated vapor, and q is the quality of the wet vapor.

The total heat of a wet vapor is the sum of the heat of the liquid and the latent heat of evaporation. This may be expressed by the equation

$$H_w = q\,L + h,$$

in which H_w is the total heat of the wet vapor, h is the heat of the liquid, and q and L are as in the preceding paragraph.

Unless the quality of a wet vapor is very low, the volume of the liquid which it contains is only a small proportion of the whole volume. We may therefore take as the specific volume of a wet vapor the product of the specific volume of the dry and saturated vapor at the same temperature (or pressure) into the quality of the wet vapor. This neglects, of course, the volume of the liquid, but no material error is introduced, as this is entirely negligible. We may then write for the specific volume of a wet vapor the formula

$$V_w = q\,V,$$

in which V_w is the specific volume of the wet vapor, q is the quality, and V is the specific volume of the dry and saturated vapor at the same temperature.

The density of a wet vapor is the reciprocal of its specific volume and is therefore equal to the density of the dry and saturated vapor at the same temperature (or pressure) divided by the quality of the wet vapor.

The external work of evaporation of a wet vapor is equal to the external work of evaporation of the dry and saturated vapor at the same temperature (or pressure), multiplied by the quality of the vapor.

The internal energy of evaporation of a wet vapor is equal to the internal energy of evaporation of the dry and saturated vapor at the same temperature (or pressure) multiplied by the quality of the vapor.

The entropy of the liquid is the same in the case of a wet vapor as in the case of the dry and saturated vapor of the same temperature (or pressure).

The entropy of evaporation of a wet vapor is equal to the entropy of evaporation of the dry and saturated vapor at the same temperature (or pressure) multiplied by the quality of the vapor.

The total entropy of a wet vapor is equal to the sum of the entropies of the liquid and of evaporation and may be expressed by the formula

$$N_w = M + \frac{qL}{T},$$

in which N_w is the total entropy of the wet vapor, q is its quality, M is the entropy of the liquid of the dry and saturated vapor, and $\frac{L}{T}$ is the entropy of evaporation of the dry and saturated vapor.

The above properties, when determined by the methods given, will of course be for 1 pound of wet vapor. The properties of the dry and saturated vapor, in the case of steam or other vapors used in thermodynamic machinery, may be taken from tables. If the quality of the wet vapor is unknown, but its temperatures or pressure, and its total heat or total entropy, or density or specific volume, or its latent heat or entropy of evaporation is known, its quality, and from this its other properties, may be computed from the equations or by the methods developed in the preceding paragraphs.

100. Superheated Vapors. When a vapor has a higher temperature than the temperature of vaporization corresponding to its pressure, it is said to be superheated. The state of a superheated vapor is defined in practice by giving either its temperature and pressure or by giving its pressure and the amount of superheat. The **amount of superheat** is obtained by subtracting from the observed or computed temperature of the superheated vapor the temperature of vaporization corresponding to the observed or computed pressure of the vapor.

101. Properties of a Superheated Vapor. The latent heat of evaporation, the temperature of vaporization, the entropy of the liquid, the entropy of evaporation, and the external and internal energy of evaporation are the same for a superheated vapor as for a dry and saturated vapor when it is of the same **pressure** as the superheated vapor.

The heat of superheat of a vapor is the quantity of heat which must be imparted to 1 pound of it in raising it from the temperature of vaporization to its actual temperature, at the pressure of vaporization. This is equal to the amount of superheat multiplied by the mean specific heat of the vapor at constant pressure, for the given conditions. It may be noted that the specific heat of a vapor at constant pressure varies both with the temperature and with the pressure, so that its mean value, for the particular range of temperature and pressure for which the computation is made, should be employed. The specific heat of superheated steam has been determined with considerable accuracy by several observers. Thomas's method consists in electrically heating steam already slightly

superheated, and measuring the energy required, the weight of steam superheated, and the rise in temperature.

The **total heat of a superheated vapor** is equal to the total heat of the dry and saturated vapor at the same pressure plus the heat of superheat. This may be expressed by the formula [1]

$$H_s = H + C_p(t_s - t)$$

in which H_s is the total heat of the superheated vapor, H is the total heat of the dry and saturated vapor at the same pressure, t_s is the temperature of the superheated vapor, t is the temperature of vaporization corresponding to its pressure, and C_p is the mean specific heat of the superheated vapor at constant pressure for the pressure and range of temperature for which the computation is made.

When the amount of superheat is not great, the specific volume of a superheated vapor may be obtained from the equation

$$V_s = \frac{V\,T_s}{T},$$

in which V is the specific volume of dry and saturated vapor of the same pressure, T is the absolute temperature of vaporization corresponding to this pressure, and T_s is the actual temperature of the vapor. In case the superheat is great, the vapor becomes more like a perfect gas in its behavior and its specific volume may be found from the characteristic equation of gases, or better, by means of an empirical equation derived from a knowledge of its actual behavior at different temperatures and pressures. The density of a superheated vapor is the reciprocal of its specific volume.

The entropy of a superheated vapor is found by adding to the entropy of the dry and saturated vapor of the same pressure the quantity obtained by a step-by-step integration of the heat additions necessary to superheat the vapor, each divided by the absolute temperature at which they occurred.

[1] When superheated steam flows through a porous plug (a process called **throttling**), it neither gains nor loses heat. Consequently we may write the formula

$$H' + C_p'\,(t_s' - t') = H'' + C_p''\,(t_s'' - t''),$$

in which the right-hand member is the total heat before throttling and the left-hand members the total heat after throttling. In each member the first term will usually be large as compared with the second, and an error in the determination of C_p will therefore have a comparatively small effect upon the answer when we solve for H' or H''. Consequently, if the total heat of dry and saturated steam be determined for some one pressure, from a series of throttling experiments, the total heats at other pressures may be determined with great accuracy. This method is due to Davis.

If the specific heat of superheat of the vapor be assumed to be constant, this quantity may be expressed by the equation,

$$N_s = N + C_p \log_e \frac{T_s}{T},$$

in which N_s is the total entropy of the superheated vapor, N is the total entropy of dry and saturated vapor of the same pressure as the superheated vapor, C_p is the specific heat of the superheated vapor at constant pressure, T_s is the absolute temperature of the superheated vapor, and T is the absolute temperature of vaporization corresponding to the pressure of the superheated vapor. In practical work, the entropy of superheated steam as well as the values of the other properties are usually obtained from a table.

102. The Relation between Vapors and Gases. At this point, it is proper to point out the relations existing between vapors and gases. It has already been stated that when a gas is sufficiently cooled and compressed, it will condense into a liquid. During the process of cooling, it is reduced from a sensibly perfect gas, first to the condition of a highly superheated vapor, then to the condition of a slightly superheated vapor, then to the condition of a wet vapor, and finally it is entirely transformed into a liquid. There is no definite line of demarcation which separates any one of these states from the next. We may therefore regard a gas as being in the condition of a highly superheated vapor even though the gas be sensibly perfect.

103. The Critical State. Experiment shows that when an attempt is made to liquefy any of the permanent gases by the application of pressure, that the attempt will fail unless the temperature of the gas is below a certain definite value. This temperature is known as the **critical temperature** of the gas. Experiment has also shown that the latent heat of evaporation of a liquid diminishes as the temperature and pressure increases, and that in the case of some liquids it is reduced to zero at the critical temperature. If a liquid is heated to the temperature at which its latent heat of evaporation becomes zero, it will, obviously, be vaporized without further addition of heat, and at any higher temperature the substance can exist only as a vapor. Consequently, the critical temperature of a substance may be defined as that temperature at which the latent heat of evaporation of its liquid becomes zero. The pressure of a saturated vapor at the critical temperature is known as the **critical pressure** of the substance. This is, of course, the pressure which is required in order to liquefy the vapor when it has the critical temperature. The specific volume of a vapor at the critical temperature and pressure is termed the **critical volume.** The state of the vapor is termed the **critical state.**

104. The Phenomena of Fusion. It is a matter of experience that when a liquid is cooled, it will finally be transformed into a solid at some definite temperature which is known as the **freezing point** of the liquid, and also as the **melting point,** or **temperature of fusion** of the solid into which it is transformed. Some complex organic substances and some mixtures of simple substances do not have a definite freezing-point, but all elementary substances do, as do also almost all simple compounds. In order to transform the liquid into a solid, it is necessary to abstract heat from it at constant temperature. In order to retransform the solid into a liquid, it is necessary to add to this the same quantity of heat, at the same constant temperature. This quantity of heat is known as the **latent heat of fusion.** The melting-point of any substance varies somewhat with the pressure, but the range is usually very narrow.

105. Sublimation. If the pressure of vaporization of a liquid at the melting-point of the solid from which it is formed is greater than atmospheric pressure, the solid cannot be melted in an open vessel, but will **sublime** at atmospheric pressure. A substance sublimes when its vapor is formed directly from its solid form by the addition of heat. When the vapor so formed is condensed, it will condense in the form of a solid.

It is evident that a solid substance can be melted only in the presence of its own vapor, and the pressure of the vapor must be equal to the saturation pressure at the temperature of fusion, for, if the vapor pressure be less than the saturation pressure, the liquid will be transformed into vapor the instant the solid melts. Hence the phenomena of sublimation. The pressure of other vapors and gases present has no effect in preventing sublimation, except as the presence of such gases serves to prevent the free escape of the vapor which is being sublimed.

An inspection of a steam table will show that, when the pressure of the water vapor present in the air is less than 0.0866 pounds per square inch, ice will sublime, since as fast as the ice is melted, the water formed will instantly disappear as vapor, the vapor pressure of water at the melting-point of ice being 0.0866 pounds per square inch. In order to obtain water from ice it is therefore necessary to melt the ice in an atmosphere where the pressure of the water vapor is greater than the value given. Carbon is an example of a substance which cannot be liquefied except at very high pressures, and since the temperature at which carbon would melt is exceedingly high, it is impossible, by any means, to obtain liquid carbon. Could we do so, it would in all probability crystallize in the form of the diamond, on solidifying, just as ice crystals are formed from water at suitable pressures (i.e., at a pressure greater than 0.0866 pounds per square inch).

Most solids sublime to a noticeable extent at temperatures approaching their melting-point; carbon, for instance, sublimes from the filament

within the bulb of the incandescent lamp, and is deposited in the form of a thin film on the interior of the glass. Ice and snow also sublime at temperatures far below freezing. Even a cold wind will cause the rapid disappearance of a snowbank provided the air is dry (i.e., the pressure of the water vapor present is very low). While we cannot state positively, it is quite possible that all solid substances are continually subliming at a very slow rate, and would in the course of centuries lose appreciably in weight. We know that this is true in the case of certain substances, which gradually disappear at temperatures below their melting-point unless confined within an air-tight space.

106. Isothermal Expansion of Vapors. When a vapor is caused to expand without the addition of heat, its temperature falls. Consequently, if a vapor is caused to expand isothermally, heat must be added to it. If the vapor be wet, this heat will vaporize the moisture present as the expansion progresses, and so long as any moisture is present, the expansion will be isobaric as well as isothermal, for the least fall in pressure will lower the temperature of vaporization of the liquid, and cause it to evaporate at such a rate as to restore the pressure to that corresponding to the temperature. For instance, if a mixture of steam and water be confined at constant pressure and heat be supplied, the water will be evaporated, and the volume will increase, but the pressure and temperature of the mass will both remain constant. The pressure-volume curve which represents the isothermal expansion of a mass of wet vapor is, therefore, a horizontal line. The work done during such isothermal expansion is, of course, equal to the product of the pressure (in pounds per square foot) into the change in volume (in cubic feet), and is also equal to the external work of evaporation of the quantity of liquid evaporated during isothermal expansion. The quantity of liquid so evaporated may be deduced from the specific volume of the vapor and the observed or computed change in volume during the expansion.

If the isothermal expansion of a vapor be continued after it has become dry, the vapor will become superheated, since the pressure will fall off and the temperature will remain constant. As the expansion progresses, the amount of superheat becomes greater and greater as the pressure becomes lower, and the condition of the vapor approaches more and more that of a perfect gas. The pressure-volume curve for the isothermal expansion of a superheated vapor resembles that of a gas and may be very nearly represented by a rectangular hyperbola. This method of vaporous expansion is not, however, of great importance in the theory of thermodynamic machinery, since it is never met with in practice.

107. Expansion without Change of Quality. A mass of vapor may be caused to expand, and to remain in the dry and saturated condition throughout the expansion, by the addition or abstraction of heat at the

proper rate. The pressure-volume curve of a mass of steam, when it expands under such circumstances, is known as the line of constant steam weight, and it may be plotted, point by point, from a steam table, by making the volume of the mass of vapor proportional to the specific volume of dry and saturated steam for each of the several pressures for which the points are plotted. The plotting of this curve is a matter of importance in the analysis of steam-engine and steam-turbine tests. If a vapor which condenses by adiabatic expansion be caused to expand slowly within a conducting cylinder while the walls of the cylinder are maintained at a temperature slightly higher than the initial temperature of the vapor, it will expand in this manner, since a wet vapor quickly takes up heat, while a dry one does not. As soon as any of the vapor is condensed by expansion, the liquid formed is immediately re-evaporated by the heat from the walls of the cylinder. In certain kinds of engineering apparatus it is found that this method of vaporous expansion is quite closely approximated.

108. Adiabatic Expansion. A vapor is caused to expand adiabatically when it is confined within a non-conducting cylinder or when it is allowed to flow through a properly formed nozzle. The successive states of a mass of vapor undergoing adiabatic expansion all have the same entropy. In the case of a vapor, we cannot write a rational equation connecting the pressure and volume, or temperature and volume, of the mass of expanding fluid, as we can in the case of a gas. It is therefore impossible to compute directly the exact effects of adiabatic expansion upon the temperature, pressure, and quality of the vapor, although numerous empirical equations have been given by different investigators which give results which are approximately correct for limited ranges of expansion. However, by means of the relations between the total entropy of a vapor and its other properties, we may compute for any particular case, the properties of the vapor when its initial temperature or pressure and quality and its final temperature are known. Thus if vapor of known properties (i.e., temperature or pressure, quality and total entropy), be caused to expand adiabatically to some other temperature or pressure, its total entropy at the new state will be the same as it was initially. From the known total entropy and temperature or pressure at the new state, we may compute the entropy of vaporization, the quality of the vapor, and any other properties which are desired. For instance, if steam of 350° temperature and 98 per cent quality be caused to expand adiabatically to a temperature of 110°, we may find its properties at the lower temperature in the following manner: The entropy of the liquid at 350° is 0.5032. The entropy of evaporation of the wet steam is $0.98 \times 1.0748 = 1.0533$. The total entropy of the steam at 110° will, since the expansion is adiabatic, be the same as it was at 350°, namely, $.5032 +$

1.0533 = 1.5565. The entropy of evaporation will be found by subtracting from the total entropy of the steam the entropy of the liquid at 110°, which is 0.1471, giving for the entropy of evaporation of the wet steam 1.4094. The quality of the steam may now be found by dividing the entropy of evaporation of the wet steam by that of dry and saturated steam. In this case the quality will be

$$\frac{1.0094}{1.8082} = 77.9 \text{ per cent.}$$

From this quality the other properties of the wet steam may be determined.

109. Effect of Adiabatic Expansion on the Properties of a Vapor. When the total entropy of dry and saturated vapor decreases as the temperature of vaporization increases, the vapor will be partly condensed as a result of adiabatic expansion unless it is highly superheated or very wet at the beginning of the expansion. Most vapors are of this character, steam being a good example of the type. When steam expands adiabatically, a portion of it will condense so long as the initial quality of the steam is greater than about 50 per cent. On the other hand, the properties of certain kinds of vapors are such that the entropy of the dry and saturated vapor increases with the temperature of vaporization. Such vapors superheat when they expand adiabatically. Ether is an example of such a vapor. If it be initially dry and saturated and be caused to expand adiabatically, it will be superheated, while if it is initially wet, it will become dryer as a result of the expansion. Vapors which condense by adiabatic expansion are dried or superheated by adiabatic compression, and those which are superheated or dried by expansion are condensed by adiabatic compression. Thus steam when very wet is condensed, and when nearly dry is dried by adiabatic compression.

We may determine the adiabatic expansion line of a vapor point by point, by determining its specific volume at several pressures by the principles outlined in Art. 108. In the same way we may determine the total heat, or any other desired property, of an expanding vapor for different temperatures (or pressures). If desirable, we may derive an empirical equation which will give the relation between the property desired and the temperature or pressure of the expanding vapor. The design of turbine nozzles and of other forms of steam machinery may be greatly facilitated by employing, in such computations, a table or diagram which gives the relations between the temperature, quality, entropy, specific volume and total heat of a vapor. Peabody's temperature-entropy table is an example, giving the relation of the quality, total heat, and specific volume of steam to its temperature and entropy. Mollier's diagram, also much used for this work, gives the relations of the

quality or superheat, and the pressure of steam, to its total heat and entropy.

110. Work of Adiabatic Expansion. The quantity of work done by a mass of vapor during adiabatic expansion will depend upon the mass and the initial quality or superheat of the vapor, and upon the temperature or pressure limits of the expansion. It will be equal to the difference between the initial and final internal energy of the vapor. For a further development of the theory of adiabatic expansion of vapors, as applied in practice in the design of steam turbines, see Arts. 201 and 202.

111. Determination of the Quality of a Wet Vapor. In practice, all vapors which are not superheated are wet, since it is impossible by any means at our command to obtain a vapor which is exactly dry and saturated, just as it is impossible to obtain two points which are exactly a given distance apart. Engineering investigations often therefore involve the determination of the quality or superheat of a vapor. It is not difficult to measure simultaneously the temperature and pressure of a superheated vapor and thereby determine the superheat, but it is necessary to resort to indirect methods in order to determine the quality of a wet vapor. The vapor whose quality engineers are most often obliged to determine is steam. An instrument for determining the quality of steam is termed a **steam calorimeter,** and several types of such instruments are in use.

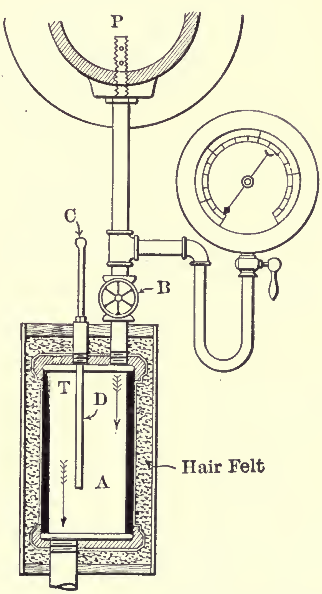


Fig. 19.—The Peabody throttling calorimeter.

112. The Throttling Calorimeter. When the steam which is being tested contains less than 3 or 4 per cent of moisture, and the pressure is sufficiently high, it is usual to employ a type of calorimeter originally devised by Professor Peabody, which is known as a throttling calorimeter, and is shown in Fig. 19. The essential parts consist of a chamber A, usually made of pipe fittings, a valve B, which is interposed between the chamber and the source of steam, and a thermometer C, which is inserted in a thermometer well D near the center of the chamber. The steam to be tested is taken from pipe P in which it is flowing and admitted to the chamber through the valve, which is kept nearly closed, so that the

pressure of the steam in the chamber is about that of the atmosphere. The pressure of the steam in the pipe P must be known. The total heat per pound is then given in the formula,

$$H_w = qL_1 + h_1, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

in which H_w is the total heat per pound of the wet steam in the pipe P, q is the quality of the steam in the pipe, L_1 is the latent heat of evaporation of dry and saturated steam having the pressure of the steam in the pipe P, and h_1 is the heat of the liquid at the pressure of the steam in the pipe. After passing through the valve the total heat of the steam will be unaltered, but since the total heat of dry and saturated steam as atmospheric pressure is less (in case q is sufficiently large) than the total heat H_w of the wet steam, the steam in chamber A will be superheated, and the amount of its superheat may be determined by means of the thermometer. The total heat of the superheated steam in A will be given by the formula,

$$H_w = H_a + .47(t - t_a), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

in which H_a is the total heat of dry and saturated steam at atmospheric pressure, t is the F. temperature registered by the thermometer, t_a is the saturation temperature of steam at atmospheric pressure, and .47 is the specific heat of superheated steam at atmospheric pressure. Equating 1 and 2 we will have

$$qL_1 + h_1 = H_a + .47(t - t_a) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Solving 3 for q we will have,

$$q = \frac{H_a - h_1 + 0.47(t - t_a)}{L_1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

113. Errors of the Throttling Calorimeter. The results given by the throttling calorimeter are affected by the following sources of error: First, loss of heat by radiation, which makes the total heat of the steam in the chamber A less than the total heat of the steam in the pipe and reduces the apparent quality of the steam; second, back pressure in the chamber A which increases the temperature registered by the thermometer and the apparent superheat in the chamber A; third, the temperature of the blast of steam issuing from the valve is less (since part of its total heat is in the form of kinetic energy) than that of the steam in the chamber A (where the kinetic energy of the blast has been retransformed into heat), hence if this blast of steam strikes the thermometer well, the thermometer reading will be lower than it should be; fourth, the sample of steam taken from the pipe may contain a greater or less proportion of water than the

steam flowing in the pipe. The first source of error may be obviated by clothing the calorimeter in some non-conducting material or by arranging it so that the chamber A is surrounded by a steam jacket, as is done in the New Hampshire calorimeter shown in Fig. 20. The second source of error may be eliminated by obtaining the exact pressure of the steam in the calorimeter by means of a pressure gage attached to the chamber A. It is usually more convenient and quite as satisfactory to allow an ample opening for the escape of steam, so that the pressure in the calorimeter shall be only a small fraction of a pound greater than the pressure of the atmosphere. The third source of error may be eliminated by so designing the calorimeter that the blast of steam from the reducing valve does not strike upon the walls of the thermometer well. The fourth source of error is the most difficult of all to eliminate. Steam rising through a vertical pipe is of practically uniform quality, hence a sample taken from such a pipe will represent accurately the quality of the steam.

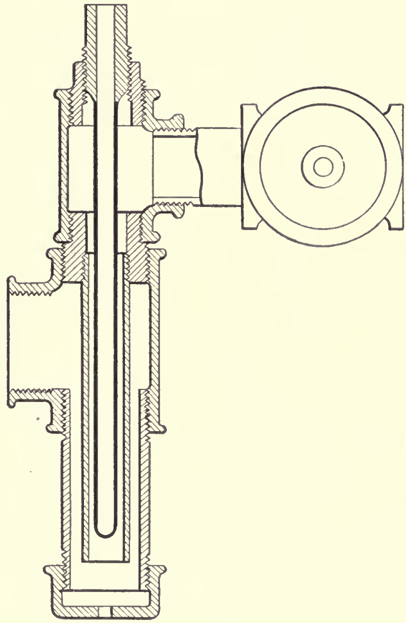


Fig. 20.—New Hampshire throttling calorimeter.

When steam flows through a horizontal pipe a quantity of water flows along the bottom of the pipe and the lower strata are wetter than the upper ones. When a sample of steam is taken from a horizontal pipe through an opening distant 45° from the bottom of the pipe, as shown in Fig. 21, it will represent the average quality of the steam in the pipe with considerable accuracy. A sample of steam from a pipe in which the steam is flowing downward may be taken by introducing into the pipe

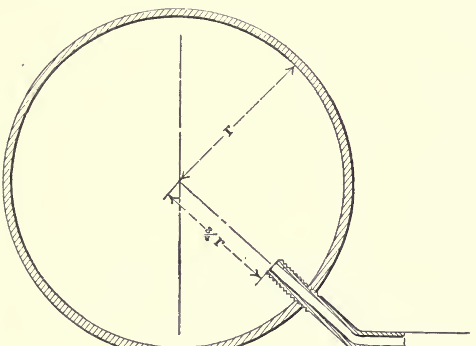


Fig. 21.—Sampling pipe for a horizontal steam pipe.

a small pipe in which are drilled a number of small holes in the manner shown in Fig. 22. It is preferable, when it is possible, to take steam in this manner from a vertical pipe and better to take it from a pipe in which the current of steam is rising rather than from one in which it is descending. Fortunately, the two points in a boiler plant where it is most necessary to make determinations of the quality of steam are the point where the steam issues from the boiler and the point where the steam enters the engine. In the first case, the steam is of uniform quality, and it is almost always possible to take the sample from a vertical pipe. In the second case, the steam is usually passed through a separator before entering the engine, and the action of the separator is such that the steam entering the engine is uniform in quality. If any suspicion exists that the steam is not uniform in quality, great pains must be taken to insure that the sample truly represents the actual quality of the steam.

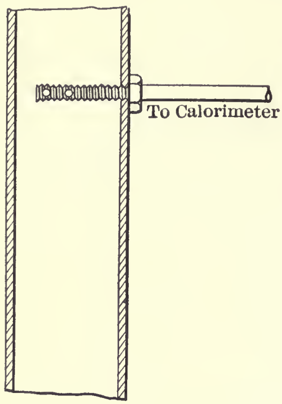


FIG. 22.—Sampling pipe for a vertical steam pipe.

114. Other Calorimeters. In case the amount of moisture in the steam is large, the quality of the steam may be determined by condensing the steam in a known weight of water and determining the rise in temperature. A calorimeter operating on this principle is known as a barrel calorimeter. Into a known weight of water in a barrel is introduced a pipe through which steam flows. The condensation of the steam raises the temperature of the water and the heat lost by the steam is equal to the heat gained by the water. Let W be the weight of water originally contained in the barrel, and w be the gain in weight or weight of steam condensed. Let t_1 be the initial temperature of the water, and t_2 be the final temperature of the water. Let h_1 and h_2 be the heat of the liquid (as obtained from a steam table) at the temperature t_1 and t_2 respectively. Let L be the latent heat of evaporation of dry and saturated steam at the pressure of the steam which is sampled, and h_3 be the heat of the liquid at this pressure. The heat gained by the water will then be $W(h_2-h_1)$ and the heat lost by the steam will be $w(qL+h_3-h_2)$ since the steam is condensed and the liquid reduced to the temperature corresponding to h_2. Equating these expressions and solving for q, we will have

$$q=\frac{W(h_2-h_1)-w(h_3-h_2)}{wL}.$$

The barrel calorimeter is not as accurate an instrument as the throttling calorimeter, but if properly used it will give fairly good results. The

accuracy of the calorimeter will obviously depend upon the accuracy of the thermometer readings, and of the weight obtained, upon a thorough stirring of the water in the barrel so that all parts are of the same temperature, and upon the length of time which the experiment takes. The shorter the time of the experiment, other things being equal, the more accurate the results will be. The barrel calorimeter usually gives results which are too low.

The separating calorimeter is an instrument which mechanically separates the water from the steam. The water is then weighed or measured and the steam is weighed or estimated in some way. The weight of steam divided by the weight of water plus the weight of steam gives the quality of the steam. Carpenter's separating calorimeter, shown in Fig. 23, is an example of this class. The steam enters the instrument at the top and issues into the body of the calorimeter through the several holes in the pipe *A*. When the direction of motion of the steam is suddenly changed, by causing the steam to pass out of the chamber *B* in an upward direction, the superior inertia of the heavy particles of water carries them against the perforated metal basket *C*, to which they adhere. The water so collected drips into the chamber *D*, where it is measured by means of the gage glass *E*. The dry steam following the path shown by the arrow escapes through an orifice at the bottom of the calorimeter. It may be shown experimentally that so long as the pressure in the calorimeter is more than twice that of the atmosphere, the weight of steam escaping through this orifice in a given time is very nearly proportional to the absolute pressure of the steam in the calorimeter. This absolute pressure is measured by the steam gage *G*. The weight of dry steam discharged in a given time is then given by the formula,

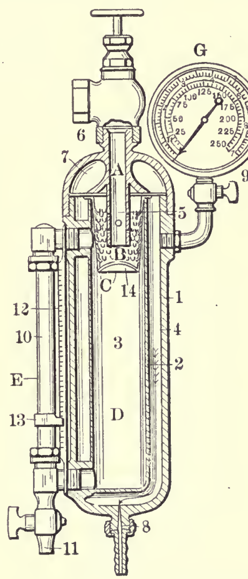


FIG. 23.—Carpenter's separating calorimeter

$$W = K t p,$$

where K is the discharge constant, t is the time (in minutes) and p is the absolute pressure in the calorimeter. K is to be determined for any particular instrument, by condensing the steam discharged at known pressure in a known time. If w be the weight of water collected in the calorimeter in time t, the quality of the steam will be

$$q = \frac{W}{W + w}.$$

The errors of the separating calorimeter are those due to radiation and those due to incomplete separation of the water from the steam. These two sources of error tend to correct one another, and the latter is usually greater in amount than the former, so that the apparent quality, as obtained by the use of the separating calorimeter, is usually greater than the true quality. The separating calorimeter is a convenient instrument to use, but its discharge constant and its errors should be determined before the instrument is used.

The quality of steam may be obtained by drying and superheating it by means of a measured quantity of electrical energy, as is done in the Thomas superheating calorimeter. Various types of steam calorimeters and the proper methods of using them are described in Carpenter's Experimental Engineering, Chapter 13.

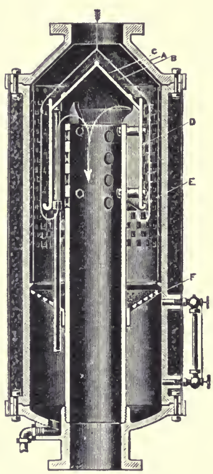


Fig. 24.

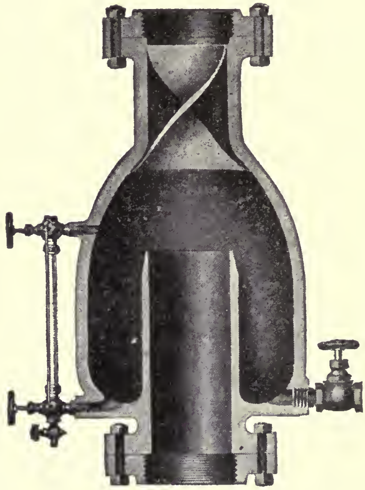


Fig. 25.

115. The Steam Separator. It is usual, when steam engines are distant from the boilers supplying them with steam, or when for any reason the steam supply is likely to be wet, to interpose in the steam pipe, close to the engine, an apparatus termed a **separator,** whose duty it is to remove from the steam the most of the water which it contains. Separators operate on two principles. In the first type the steam passes into the separator at high velocity and its direction is suddenly changed. The wet steam consists a mixture of vapor and water. The particles of water being many hundred times heavier than the steam, their superior inertia will cause them to shoot straight ahead when the current of steam is suddenly deflected. If these particles of water encounter a wet surface, they will adhere to the surface and drip to the bottom of the chamber in which it is contained. Such a separator is shown in Fig. 24.

In the second type of steam separator, the steam is caused to travel in a helical path; the centrifugal force so developed, on account of the superior density of the water particles, throws them against the outside walls of the separator, upon which they collect. The water then drips to the bottom of the separator. Such a separator is illustrated in Fig. 25.

A separator will usually, in case very wet steam is supplied to it, deliver steam of 98 per cent quality or better. In case the steam is fairly dry (i.e., of 96 per cent quality or better) the greater part of the moisture present will be extracted by the separator. Steam containing less than 2 per cent of moisture is usually known as commercially dry steam. Such steam may be always obtained by the use of a suitable separator. The water which collects in a separator must be drawn away at intervals, for, if the separator fills up with water, it is no longer effective. The water from a separator is usually taken care of by an automatic device termed a steam trap, which discharges the water flowing into it, at intervals, without permitting the escape of steam.

PROBLEMS

1. How much water is contained in 8 lbs. of wet steam whose quality is 85%?
Ans. 1.2 lbs.

2. What weight of dry and saturated steam is contained in 10 lbs. of wet steam having 15 per cent of moisture? Ans. 8.5 lbs.

3. What is the heat of the liquid of wet steam at a pressure of 1 atmosphere?
Ans. 180 B.T.U.

4. What is the latent heat of evaporation of 1 lb. of steam of 90 per cent quality at a pressure of 1 atmosphere? Ans. 873.4 B.T.U.

5. What is the total heat of 1 lb. of steam of 90 per cent quality at a pressure of 1 atmosphere? Ans. 1053.4 B.T.U.

6. What is the specific volume of steam of 90 per cent quality at a pressure of 1 atmosphere? Ans. 26.93 cu.ft.

7. What is the density of steam of 90 per cent quality at a pressure of 1 atmosphere?
Ans. 0.0416 lbs. per cu.ft.

8. What is the external work of evaporation of steam of 90 per cent quality at a pressure of 1 atmosphere? Ans. 65.5 B.T.U.

9. What is the internal energy of evaporation of steam of 90 per cent quality at a pressure of 1 atmosphere? Ans. 807.8 B.T.U.

10. What is the total internal energy of steam of 90 per cent quality at a pressure of 1 atmosphere? Ans. 987.8 B.T.U.

11. What is the entropy of the liquid of steam of 90 per cent quality at a pressure of 1 atmosphere? Ans. 0.3118.

12. What is the entropy of evaporation of steam of 90 per cent quality at a pressure of 1 atmosphere? Ans. 1.3003.

13. What is the total entropy of steam of 90 per cent quality at a pressure of 1 atmosphere? Ans. 1.6121.

14. The latent heat of evaporation of wet steam for a temperature of 300° is 800 B.T.U. What is its quality? Ans. 88%.

15. The total heat of wet steam at a pressure of 100 lbs. is 1100 B.T.U. What is its quality? Ans. 90.3%.

16. Two pounds of wet steam are contained in a volume of 12.96 cu.ft. at a temperature of 280°. What is the quality? Ans. 75%.

17. Wet steam is found to weigh 0.120 lbs. per cubic foot at a pressure of 50 lbs. What is its quality? Ans. 98%.

18. The entropy of evaporation of wet steam at a temperature of 180° is 1.400. What is its quality? Ans. 90.5%.

19. The total entropy of wet steam at a temperature of 330° is 1.400. What is its quality? Ans. 82.1.%.

20. Steam having a pressure of 150 lbs. per square inch has a temperature of 408.5°. What is the amount of superheat? Ans. 50°.

21. Assuming the specific heat of superheated steam at atmospheric pressure to be 0.47, what will be the heat of superheat of 1 lb. of steam at a temperature of 312° and a pressure of 1 atmosphere? Ans. 47 B.T.U.

22. What will be the total heat of this steam? Ans. 1197.4 B.T.U.

23. From a table of the properties of superheated steam find the specific volume, the total heat, and the entropy of steam of 100 lbs. pressure and having a temperature of 427.8°. Ans. 5.14 cu.ft., 1239.7 B.T.U., and 1.6658.

24. From such a table determine the temperature of steam whose pressure is 120 lbs. and whose total heat is 1233.8 B.T.U. Ans. 421.3°.

25. Determine the number of degrees of superheat of steam of 150 lbs. pressure whose entropy is 1.6343. Ans. 100°.

26. What quantity of heat is required to raise 1 lb. of water from a temperature of 60° to a temperature of 212°? Ans. 151.92 B.T.U.

27. What quantity of heat is required to evaporate 1 lb. of water of a temperature of 60° into dry and saturated steam at a temperature of 212°? Ans. 1122.3 B.T.U.

28. What quantity of heat will be required to evaporate 10 lbs. of water having a temperature of 80° into steam of 90 per cent quality at a pressure of 100 lbs. absolute? Ans. 10,423 B.T.U.

29. What quantity of heat will be required to evaporate 100 lbs. of water of a temperature of 60° into steam at a pressure of 150 lbs. and a temperature of 448.5°? Ans. 121,630 B.T.U.

30. One pound of steam of a pressure of 100 lbs. per square inch and a quality of 50 per cent is expanded isothermally until it is dry and saturated. Find the work done and the heat supplied. Ans. 31,890 ft.-lbs. and 444.0 B.T.U.

31. Steam having a volume of 2 cu.ft. and a pressure of 100 lbs. is expanded while it remains in a dry and saturated condition to a pressure of 10 lbs. What is its final volume? Ans. 17.33 cu.ft.

32. Dry and saturated steam at a pressure of 100 lbs. expands adiabatically until its pressure is 10 lbs. What is its final total entropy, entropy of evaporation, and quality? Ans. 1.6002, 1.3188, and 87.67%.

33. What will be the total heat of the steam when the expansion has proceeded to 2 lbs. absolute? Ans. 930.0 B.T.U.

34. Steam having a quality of 10 per cent is compressed adiabatically from a pressure of 10 lbs. absolute to a pressure of 30 lbs. absolute. What is its quality? Ans. 4.9%.

35. At what pressure will this steam become completely transformed into water? Ans. 64.4 lbs.

36. What quantity of work was performed in compressing this wet steam to a pressure of 64.4 lbs.? Ans. 14.6 B.T.U.

37. What quantity of work is performed by 1 lb. of steam in expanding adiabatically from a temperature of 300° to a temperature of 100°, the steam being initially dry and saturated? Ans. 239.1 B.T.U.

38. Steam in a throttling calorimeter has a temperature of 312° at a pressure of 1 atmosphere. What is its total heat? Ans. 1197.4 B.T.U.

39. The original pressure of this steam was 200 lbs. per square inch. What was its quality before entering the calorimeter? Ans. 99.9%.

40. The temperature of the steam in a throttling calorimeter is 220° when the barometer indicates a pressure of 13 lbs. per square inch. The original pressure of the steam was 87 lbs. gage. Find the quality of the steam. Ans. 96.91%.

41. A barrel contains 400 lbs. of water at a temperature of 75° F. After condensing steam at a pressure of 120 lbs. absolute, it gains 20 lbs. in weight and 53° in temperature. Find the quality of the steam. Ans. 95.8%.

CHAPTER VII

MIXTURES OF GASES AND VAPORS

116. Gaseous Mixtures. When two or more sensibly perfect gases are brought into contact with one another, the particles of one tend to pass between the particles of the other. As a result, after the lapse of a sufficient length of time, the two gases will form a homogeneous mechanical mixture. This process of mixing is known as diffusion, and the mixture resulting is, in every sense except a chemical one, a perfectly homogeneous body, and will remain so provided it undergoes no chemical action, and its component particles are not separated by enclosing the mixture within a porous vessel.[1] When such a mixture of gases occurs, all of the constituents of the mixture will come to the same temperature, and this temperature will, of course, be the temperature of the mixture. The mass of the mixture will be the sum of the masses of the several constituents. If the mixture be confined within a vessel, each of the constituents will exert upon the walls a pressure which is equal in amount to the pressure which that constituent would exert were it confined separately within the vessel, at the same temperature. Consequently, the pressure of the mixture will be the sum of the pressures of the several constituents. Were each of the constituents confined at the pressure and the temperature of the mixture, they would occupy definite volumes. The sum of these volumes will, if the constituents are all sensibly perfect gases, be the volume of the mixture.

117. The Thermal Properties of Gaseous Mixtures. The density of such a mixture, under given conditions, may be obtained by dividing the mass of the mixture by its volume, or if the masses of the several constituents are known, by dividing the sum of the products of the mass and density of each of the constituents, under the given conditions, by the mass of the mixture. From the density of the mixture the value of R for the mixture may be deduced by the method given in Art. 30. In the same way, the specific heat of such a mixture at constant pressure (or constant volume) may be found by dividing the sum of the products of the mass and specific heat at constant pressure (or constant volume) of each of the constituents, by the mass of the mixture. The value of

[1] The process of separating the constituents of a gaseous mixture by the action of a porous wall is termed osmosis. For the theory of this action see Chapter XXVI.

the constant γ for a mixture of gases is the ratio of the specific heats at constant pressure and constant volume, and the mixture will, during expansion or compression, behave exactly as if it were a simple gas whose thermodynamic properties are those obtained in the manner given above. We may, therefore, treat such a mixture as if it were a gas having definite and known thermodynamic properties.

The mathematical statement of the properties of two or more gases may be obtained as follows: Let W' and W'' be the mass of the two constituents, R' and R'' the value of the function R for these constituents, D' and D'' the density of these constituents under standard conditions, K'_p and K''_p the specific heat at constant pressure, and K'_v and K''_v the specific heat at constant volume of the two constituents. We will then have the following relations:

The mass of the mixture will be

$$W = W' + W''. \qquad (1)$$

When the mixture is within the volume V at the temperature T, the pressure of the mixture will be

$$P = \frac{T}{V}(W'R' + W''R''). \qquad (2)$$

The density of the mixture under standard conditions will be

$$D = \frac{W'D' + W''D''}{W' + W''}. \qquad (3)$$

The specific heat of the mixture at constant volume will be

$$K_v = \frac{K_v'W' + K_v''W''}{W' + W''}. \qquad (4)$$

For the specific heat of the mixture at constant pressure, we will have the value

$$K_p = \frac{K_p'W' + K_p''W''}{W' + W''}. \qquad (5)$$

The ratio of these two quantities gives the value of γ for the mixture, which becomes

$$\gamma = \frac{K_p'W' + K_p''W''}{K_v'W' + K_v''W''}. \qquad (6)$$

The value of R for the mixture will of course be equal to

$$R = K_p - K_v, \qquad (7)$$

or

$$R = \frac{K_p'W' + K_p''W''}{W' + W''} - \frac{K_v'W' + K_v''W''}{W' + W''}, \qquad (8)$$

which becomes

$$R = \frac{W'(K_p' - K_v') + W''(K_p'' - K_v'')}{W' + W''}, \quad . \quad . \quad . \quad . \quad (9)$$

which is of course

$$R = \frac{R'W' + R''W''}{W' + W''}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

An inspection of equation (6) will show that the value of γ for the mixture of gases is not the weighted mean of the values of this constant for the constituents of the mixture, as might be supposed, from analogy with the other quantities. When there are more than two constituents present in the mixture, its properties may be determined by properly modifying the above equations. For instance, for a mixture of three gases the value of R becomes

$$R = \frac{R'W' + R''W'' + R'''W'''}{W' + W'' + W'''}.$$

118. Mixture of a Gas with a Vapor. A mixture of a gas and a vapor becomes homogeneous through diffusion in exactly the same way as do mixtures of two or more gases. The pressure exerted by the mixture upon the walls of the containing vessel is the sum of the pressures which would be exerted by the gas and the vapor were each one contained separately in the vessel at the given temperature. If the vapor be saturated on account of the presence of its liquid, the pressure which it exerts will depend upon the temperature, while the pressure which the gas will exert will depend upon its volume and mass as well as on the temperature. We may then compute the pressure exerted by the gas by subtracting from the pressure of the mixture the saturation pressure of the vapor at the temperature of the mixture. In case the vapor is superheated, the pressure exerted will, as before, be the sum of the pressures of each constituent of the mixture, but the pressure of the superheated vapor depends, like that of the gas, upon the mass of the vapor and the volume in which it is confined, and the pressure of the vapor may be obtained only by the methods described in Art. 101.

119. Air a Mixture of Gases and Water Vapor. The thermal properties of air, as given in Chapter II., are those of dry air, free from carbon dioxide. Outdoor air usually contains a minute percentage (.03 per cent) of carbon dioxide and a variable quantity of water vapor, so that the density, specific heat, and other thermal properties of the atmosphere are continually varying. Ordinarily, the properties of air may be assumed to be those of dry air free from carbon dioxide, without sensible error. In certain classes of engineering computations, however, it is essential to take account of the variation in the properties of air when varying quantities of water vapor are present. In such computations it is necessary

to consider air as a mixture of a gas of known properties, with superheated water vapor or steam.

120. Humidity of Air. When the pressure of the water vapor present in the air is the saturation pressure of water vapor at the temperature of the air, the air is said to be saturated. If the pressure is less than this quantity, the air is said to have a certain humidity, which is expressed as a per cent and is found by dividing the actual pressure of the water vapor in the air by the saturation pressure of water vapor at the temperature of the air. In case the air is saturated, it contains all the moisture which it is possible for it to hold, and any reduction in temperature will precipitate some of the moisture in the form of dew or rain. In case the humidity of the air is less than 100 per cent, the water vapor present in the air is superheated, since its temperature is greater than the temperature corresponding to its pressure, and the air may be treated as a mixture of a gas and a superheated vapor.

121. Dew Point. If a cold metallic surface be exposed to moist air, dew will gather upon it, if its temperature is less than the saturation temperature of the water vapor present in the air. The maximum temperature at which moisture will appear upon such a surface is known as the dew point. If the dew point be determined, we may, from a steam table, find the pressure of the water vapor present in the air. The difference between the dew point and the air temperature is the superheat of the water vapor present in the air. From the pressure and superheat of this vapor we may determine its density, which will be the weight of water vapor present per cubic foot of air.

122. The Wet-bulb Hygrometer. The humidity of the air is usually determined by means of an apparatus called a wet-bulb hygrometer. This instrument consists of two thermometers both protected from the direct radiation of the sun or other objects reflecting heat. The bulb of one of these thermometers is surrounded by air. The bulb of the other is enveloped in lamp wicking which dips into a cup of water. This thermometer must be so situated that the air has free access to the wicking, but must not be exposed to wind. The water contained in the wicking, being in contact with the air, will evaporate and more will be drawn up to take its place. In evaporating, the water appropriates the sensible heat of surrounding objects and reduces the temperature of the thermometer bulb below that of the air. From the difference in temperature indicated by the dry bulb-thermometer, the dew-point and the relative humidity may be computed. Since the wet-bulb thermometer receives heat by radiation from surrounding objects and the pressure of the water vapor in its neighborhood, due to the continual evaporation, is higher than elsewhere, it will not indicate a temperature as low as the dew-point. It is necessary, therefore, to make use of the following empirical table in

determining the humidity of the air from the indications of the wet-bulb thermometer:

RELATIVE HUMIDITY, PER CENT

Dry Thermometer, Deg. F	Difference between the Dry and the Wet Thermometers, Deg. F.																										
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	26	28	30
	Relative Humidity, Saturation being 100. (Barometer = 30 ins.)																										
32	89	79	69	59	49	39	30	20	11	2																	
40	92	83	75	68	60	52	45	37	29	23	15	7	0														
50	93	87	80	74	67	61	55	49	43	38	32	27	21	16	11	5	0										
60	94	89	83	78	73	68	63	58	53	48	43	39	34	30	26	21	17	13	9	5	1						
70	95	90	86	81	77	72	68	64	59	55	51	48	44	40	36	33	29	25	22	19	15	12	9	6			
80	96	91	87	83	79	75	72	68	64	61	57	54	50	47	44	41	38	35	32	29	26	23	20	18	12	7	
90	96	92	89	85	81	78	74	71	68	65	61	58	55	52	49	47	44	41	39	36	34	31	29	26	22	17	13
100	96	93	89	86	83	80	77	73	70	68	65	62	59	56	54	51	49	46	44	41	39	37	35	33	28	24	21
110	97	93	90	87	84	81	78	75	73	70	67	65	62	60	57	55	52	50	48	46	44	42	40	38	34	30	26
120	97	94	91	88	85	82	80	77	74	72	69	67	65	62	60	58	55	53	51	49	47	45	43	41	38	34	31
140	97	95	92	89	87	84	82	79	77	75	73	70	68	66	64	62	60	58	56	54	53	51	49	47	44	41	38

From Kent's "Pocket Book," 1910 Edition.

123. Normal Humidity of the Atmosphere. Except in very arid regions, the humidity of the air varies from 40 to 80 per cent, usually ranging from 60 to 70 per cent in inland districts and from 70 to 80 per cent by the sea. Consequently, air is always ready to absorb moisture, and when water is exposed to the action of the air it will be evaporated. The rate of this evaporation will depend upon the humidity and temperature of the air, and upon the amount of wind. By warming air, its humidity will be greatly diminished and its power to absorb moisture correspondingly increased. Drying kilns, cooling towers, and other forms of engineering apparatus depend upon these principles for their operation.

124. The Computation of the Properties of Moist Air. When the relative humidity is known, the pressure of the water vapor present in the air may be found by multiplying the saturation pressure of water vapor at the temperature of the air, as obtained from a steam table, by the relative humidity. The saturation temperature corresponding to this pressure is the dew-point. Conversely the humidity may be found by dividing the saturation pressure at the temperature of the dew-point by the saturation pressure at the temperature of the air. The density of the water vapor in the air may be found by multiplying the density of water vapor at the dew-point by the absolute temperature of the dew-

point, and divided by the absolute temperature of the air. Almost exactly the same result will be obtained by multiplying the density of water vapor at the temperature of the air by the relative humidity, and since this method is both very exact and very convenient, it is proper to employ it in engineering computations.

The pressure of the dry air present in the atmosphere may be found by subtracting the pressure of the water vapor from the atmospheric pressure indicated by the barometer. The density of the dry air (i.e., its weight per cubic foot) may be found from its absolute pressure and temperature by means of the characteristic equation of gases, $PV = WRT$. Adding together the density of the water vapor and the density of the dry air, we will have the weight of the atmosphere in pounds per cubic foot.

The principles which have been developed in the preceding paragraphs with regard to the pressure, density, etc., of the constituents of the atmosphere may be applied, with equal propriety, in the case of any mixture of gas and vapor.

PROBLEMS

1. A volume of 10 cu.ft. contains 1 lb. of hydrogen and 2 lbs. of nitrogen at a temperature of 550° absolute. Find the pressure of the hydrogen, of the nitrogen, and of the mixture.

Ans. 42,400 lbs. per square foot, 6060 lbs. per square foot, and 48,460 lbs. per square foot.

2. One pound of carbon monoxide and 1 lb. of marsh-gas are together contained in a volume of 4 cu.ft. at a pressure of 100 lbs. per square inch. Find the pressure of each constituent. Ans. 36.3 lbs. per square inch and 63.7 lbs. per square inch.

3. Find the density of the mixture under standard conditions.

Ans. 0.0613 lbs. per cubic foot.

4. Find the specific heat of the mixture at constant volume. Ans. 0.3202.

5. Find the value of the constant R for the mixture. Ans. 75.79.

6. Find the value of the constant γ for the mixture. Ans. 1.304.

7. A mixture of air and saturated water vapor is contained in a confined space and has a temperature of 60°. The pressure of the air is 1 atmosphere. Find the pressure of the mixture. Ans. 14.952 lbs. per square inch.

8. A mixture of air and saturated water vapor in a confined space has a temperature of 80° F. and a pressure of 1 lb. per square inch absolute. Find the pressure of the air. Ans. 0.495 lbs. per square inch.

9. If water is present and the tempreature of the mixture is increased to 200° F., what will be the pressure of the air of the water vapor, and of the mixture?

Ans. 0.605 lbs., 11.52 lbs., and 12.125 lbs.

10. The temperature of the air is 70°, the dew-point is found by experiment to be 50°, find the humidity. Ans. 49.1%.

11. What quantity of water vapor will be contained in each cubic feet of air at the above humidity? Ans. 0.000564 lbs.

12. One thousand cubic feet of air at a temperature of 60° and a humidity of 70 per cent are compressed into a volume of 200 cu.ft. What weight of moisture did the air contain before compression? Ans. 0.58 lbs.

13. What weight of moisture will the air contain at the same temperature and 100 per cent humidity after compression? Ans. 0.1656 lbs.

14. What quantity of water will be precipitated by the compression? Ans. 0.414 lbs.

15. A wet-bulb hygrometer gives readings of 75° and 68°. What is the humidity? Ans. 70%.

16. If the pressure indicated by the barometer is 14.40 lbs. absolute, what is the pressure of the dry air? Ans. 14.1 lbs. per square inch.

17. If the temperature is raised to 150° and the humidity to 100 per cent, find the final volume of the air in terms of the original volume. Ans. 1.506.

CHAPTER VIII

THE STEAM ENGINE

125. The Mechanism of the Steam Engine. A **steam-power plant** consists of a boiler for the generation of steam, an engine for the partial transformation of the heat of the steam into mechanical energy, and a condenser into which the waste steam is discharged, together with necessary auxiliary apparatus. The place of the condenser may be taken by the atmosphere, the steam being discharged into the air against the barometric pressure. Fig. 26 shows the steam engine of such a power plant in section, the engine shown being equipped with what are termed **Corliss valves.** Various other types of valves are in use for the distribution of steam to the cylinder, but the action of the engine is most readily understood when the valves are of the type shown. In the figure, the steam pipe A carries steam from a boiler to the engine. In this pipe is placed the **throttle valve** B, which is for the purpose of shutting off steam when the engine is not running. When this valve is open, steam flows from the pipe into the steam chest C. Leading from the steam chests are two ports, one to each end of the cylinder D. These ports are closed by two valves e and e', which are known as the **inlet valves.** Within the cylinder the piston F slides back and forth, being propelled alternately in each direction by the pressure of the steam. A movement of the piston from one end of the cylinder to the other is termed a **stroke.** Two successive strokes make one revolution of the engine. The total distance traversed by any point in the piston during a stroke is called the **length of stroke,** or piston travel. The **piston rod** G, which is fastened to the piston, transmits the force exerted upon the piston to the **cross-head** H, whence it is transmitted by the **connecting rod** I to the **crank** J, which is keyed to the **shaft** K. Upon this shaft is fixed a **fly-wheel,** and the shaft revolves in two or more bearings. As the piston is pushed back and forth by the steam, the intermediate mechanism pushes the crank forward and pulls it back, causing the shaft to revolve. The function of the fly-wheel is to make the rate of rotation as uniform as possible by the inertia of its revolving mass, and to carry the engine over the " dead points " which occur when the crank and connecting rod are in line at either end of the stroke, at which time the force exerted by the steam has no tendency to turn the

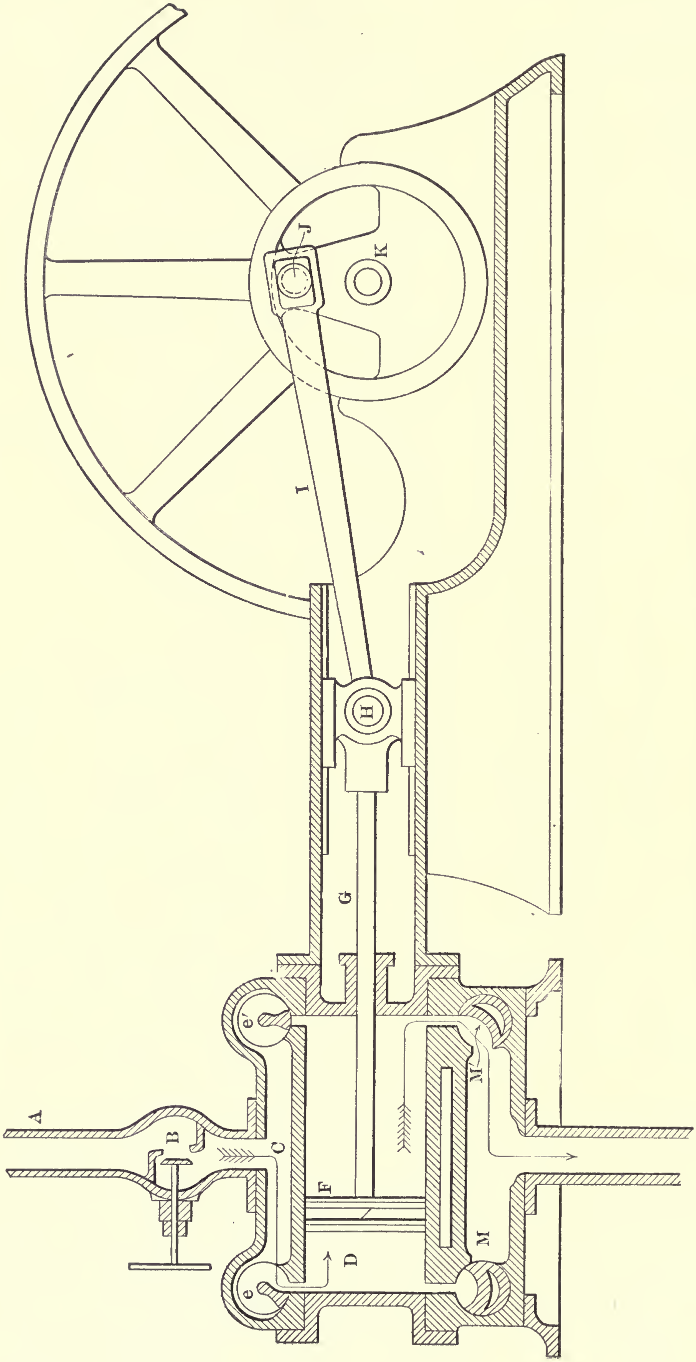


FIG. 26.—Section of a steam engine with Corliss valves.

crank. The piston is a cylindrical body, and upon the outside of this cylinder are cut grooves into which are fitted **piston rings,** whose function it is to expand against the side of the cylinder and prevent the escape of steam past the piston. The cross-head is restrained by the frame of the engine and compelled to move in a direction parallel to the axis of the cylinder. A port termed an **exhaust port** leads from either end of the cylinder into the **exhaust pipe.** These ports are closed by the exhaust valves M and M'.

126. Cycle of Operations. Assume that the various parts of the engine are each in the position shown in the drawing, the inlet valve e and the exhaust valve M' being open. Steam will enter the left-hand end (known as the **head end**) of the cylinder, and will exert a pressure upon the piston whose total amount is proportional to the product of the absolute steam pressure into the area of the piston. Since this pressure is greater than the pressure upon the opposite face of the piston (which is equal only to the pressure in the exhaust pipe) the piston will be forced to the right, rotating the crank in a clockwise direction. Steam from the boiler will flow into the head end of the cylinder, maintaining the pressure, and the steam contained in the opposite or **crank end** of the cylinder will escape through the exhaust port into the exhaust pipe. After the piston has moved forward a certain amount (usually from $^1/_6$ to $^1/_2$ of its total travel) the inlet valve e is closed by the mechanism which operates it, the exhaust valve M' remaining open. The steam which is contained within the head end of the cylinder will now begin to expand, increasing in volume and diminishing in pressure, and this expansion will continue until the exhaust valve is opened at or just before the end of the stroke. At some point near the end of the stroke, which is known as the **forward stroke,** the exhaust valve M' is closed, and just at the end of the stroke the exhaust valve M is opened. The pressure in the head of the cylinder now drops to the same value as the pressure in the exhaust pipe, the steam escaping through valve M. The pressure in the right-hand or **crank end** of the cylinder begins to rise on account of the compression of the contained steam, as soon as the valve M' is closed, and at the end of the stroke, on account of the introduction of steam from the boiler through valve e', it rises to boiler pressure. The pressure upon the right-hand face of the piston now drives it to the left, causing it to make what is termed the **return stroke.** At the proper point of the stroke the steam supply is cut off by the closing of valve e' and the steam allowed to expand, exactly as in the head end of the cylinder. As the piston approaches the head end of the cylinder, the valve M closes, the valve M' opens, and when the crank reaches **dead center** (i.e., when the connecting rod and crank are in the line and the piston has reached the limit of its travel) the valve e opens.

127. Efficiency of the Engine. Each succeeding revolution is a repetition of the events of the preceding one. At the beginning of each **working stroke** a definite amount of steam flows into one end of the cylinder from the boiler, and during the **back stroke** (or **exhaust stroke,** as it is sometimes called) the same weight of steam escapes from that end of the cylinder into the exhaust pipe. This quantity of steam is called the **cylinder feed per stroke,** or simply the **cylinder feed.** The weight of steam contained in the cylinder and clearance space at the instant the exhaust valve closes is called the **cushion steam.** The working stroke of the head of the cylinder is the back stroke of the crank end, and vice versa. The heat supplied to the engine in a given time is equal to the heat imparted in the boiler to the steam which is used by the engine during that time. The heat rejected by the engine is the latent heat of the steam (which is usually quite wet) which is rejected into the exhaust pipe during the same period. The difference between these two quantities is equal to the heat radiated from the engine during this period plus the work done by the engine during the same period. The efficiency of the engine is of course equal to the work divided by the heat supplied. It is apparent that water must be supplied to the boiler to take the place of that which is evaporated by the boiler and sent to the engine. The water so supplied is termed the **feed-water.** This water may and should have almost the same temperature as the exhaust steam which the engine rejects, since this exhaust steam may be made to surround tubes through which the feed-water is forced on its way to the boiler, or some other method of heating the feed-water by the exhaust steam may be employed. The heat supplied by the boiler to each pound of steam is then equal to the total heat of the steam (corrected, if necessary, for wetness or superheat) minus the heat of each pound of the feed-water, which is at the temperature of the engine exhaust. The heat rejected per pound of steam is the latent heat of the wet steam which is rejected. The generation of steam, of course, costs money, both for the fuel which is burned, for the labor necessary in burning it and caring for the boiler, and for other necessary expenses of operating the boiler plant. It is therefore highly desirable that the engine should be as economical as possible in the use of steam, or in other words, that it should have the highest possible efficiency.

128. Types of Engines. The engine described in the preceding paragraphs is known as a **simple non-condensing engine** when the exhaust steam is rejected into the air. If the exhaust steam is allowed to flow into the condenser or chamber in which it is cooled and condensed, and in which a vacuum pump maintains a low absolute pressure, the engine is said to be a **condensing engine.** Such an engine is usually more efficient than a non-condensing engine.

It has been found that it is both more convenient and more economical to allow steam to pass through two or more cylinders in succession, in case the steam pressure is high and the engine large. The first cylinder, instead of exhausting into the air or into a condenser, exhausts into a closed space which is known as a **receiver.** The steam from this receiver flows into a second cylinder, in which it does work, and is finally rejected to the air, or more usually to a condenser. Sometimes it is exhausted into a second receiver and flows from this into a third cylinder and occasionally into a third receiver, and thence to a fourth cylinder, before entering the condenser.

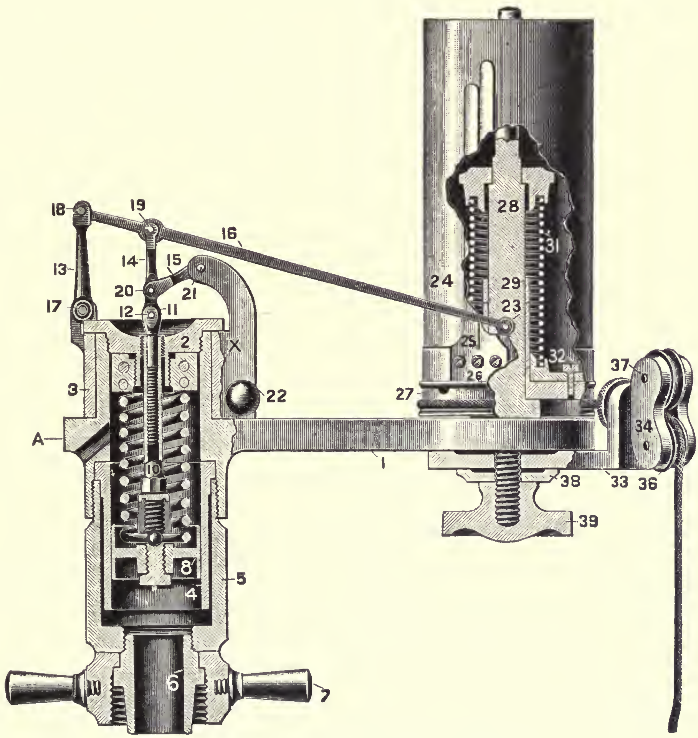


Fig. 27.—Section of an indicator.

When the steam flows through two cylinders in succession the engine is said to be **compound.** When it flows through three cylinders in succession, the engine is said to be **triple expansion,** and when it flows through four cylinders in succession, the engine is said to be **quadruple expansion.**

129. The Indicator. The pressure-volume diagram of the working fluid of an actual steam engine or other thermodynamic machine, is termed an **indicator card.** An indicator card is obtained from an engine

by the use of an instrument termed an **indicator.** An instrument of this kind is shown in section, in Fig. 27. It consists of a hollow cylinder in which is a piston which fits rather loosely, so as to move without friction. This piston is attached to one end of a small helical spring, the other end of which is fixed to the upper head of the cylinder. The pressure of the steam on the piston forces it upward against the resistance of the spring, and the amount by which the piston rises will be strictly proportional to the steam pressure producing the rise. The motion of the piston is transmitted by a rod to the parallel motion which moves

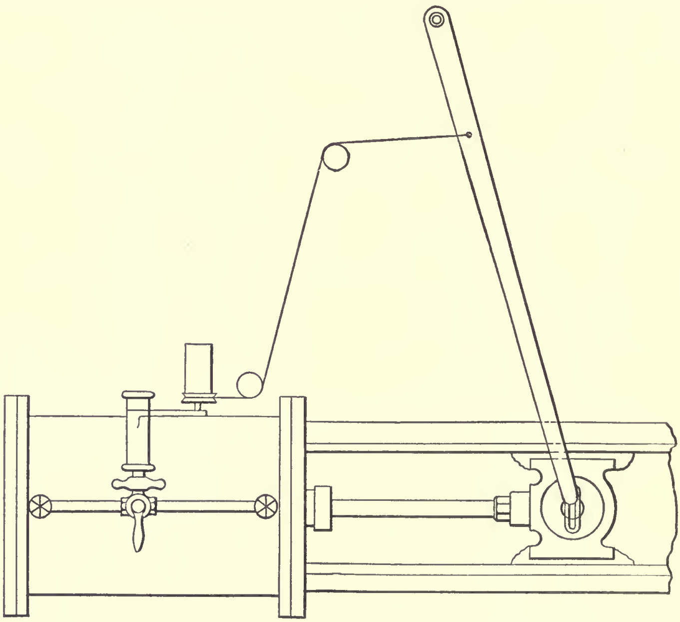


FIG. 28.—Indicator reducing motion.

the pencil point. The motion of the pencil point will then be strictly proportional to the steam pressure exerted upon the piston. The pressure, in pounds per square inch required to raise the pencil point a distance of one inch, is termed the **scale of the spring.** This pencil point is pressed lightly against a piece of paper which is wrapped about a cylinder or drum which oscillates upon its axis in unison with the motion of the piston of the engine. This is accomplished by wrapping a cord about the drum and attaching the cord to a pantograph or other reducing motion which is in turn attached to the cross-head of the engine in the manner shown in Fig. 28. The cord is drawn back during the return stroke of the engine by the action of the drum spring. The motion of the drum will then be strictly proportional to the motion of the piston, and the distance which the

drum revolves during any portion of a stroke will be proportional to the volume swept by the piston during the same time. Such an apparatus will obviously draw the *PV* diagram of the working fluid of the engine. Usually an indicator card is taken from each end of each cylinder of an engine.

130. The Theoretical Indicator Card. The indicator card which is given by a simple engine has in theory the form shown in Fig. 29. Assume that this card represents the pressure-volume diagram of the head end of the cylinder. The line OX is the line of zero pressure and the line OY, the line of zero volume, the two forming the axes of the diagram. The abscissa to a represents the volume of the working fluid contained in the cylinder when the piston of the engine is at the extreme left of its stroke. This volume is formed by the space included between the face of the pis-

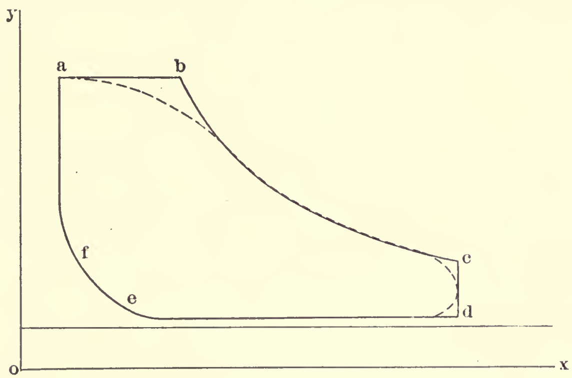


FIG. 29.—Theoretical indicator card.

ton and the head of the cylinder, the volume of the steam and exhaust ports, and the small volume in a space termed the counterbore. The distance between the face of the piston and the cylinder head, or plate which covers the end of the cylinder, is termed the mechanical clearance, and varies from $^1/_8$ to $^3/_8$ of an inch, according to the size and workmanship of the engine. In order to avoid forming a shoulder in the cylinder where the piston stops at the end of its travel, the cylinder is bored out from $^1/_8$ to $^1/_4$ of an inch larger in diameter for a short distance at either end. The space so formed is termed the **counterbore.** The sum of these three volumes taken together is termed the **clearance volume** of the engine and is usually expressed as a per cent of the swept volume of the cylinder. The **swept volume** of the cylinder is the product of the net piston area into the length of stroke. The area of the piston rod must be deducted for the crank end of the cylinder. The volume of the steam contained in the cylinder is never, of course, less than the clearance volume. As the piston moves to the right, steam follows into the cylinder from the

boiler, maintaining the pressure constant, the increase in volume meanwhile being strictly proportional to the distance which the piston travels. Point b represents the point in the stroke, and the total volume of the steam in the cylinder, at the instant that the inlet valve is shut. This is termed the point of cut-off. From this point, as the volume continues to increase with the advance of the piston, the pressure falls off, the relation between the volume and pressure being expressed very nearly, for most engines, by the equation

$$PV=K,$$

in which K is a constant equal to the product of the absolute pressure and volume at the point of cut-off. The line bc is then very nearly a

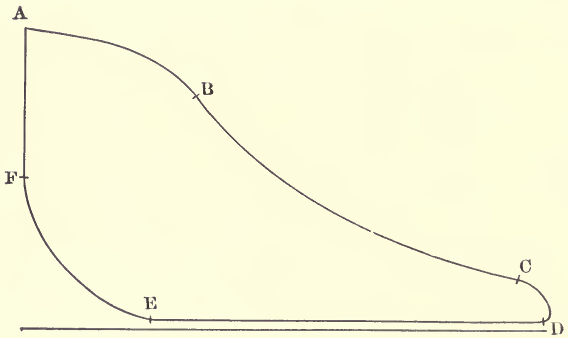


FIG. 30.—Actual indicator card.

rectangular hyperbola, having for its asymptotes the lines $O–X$ and $O–Y$. At the end of the stroke when the exhaust valve opens the pressure suddenly drops to d, the ordinate to d representing the pressure in the exhaust pipe. During the back stroke, the pressure continues at this value until the point e is reached, when the exhaust valve closes and compression commences. The compression of the mass of steam confined within the volume represented by the abscissa to e raises the pressure and also the temperature of this steam, the process of compression being represented by the line ef, which is also like the expansion line bc, very nearly a rectangular hyperbola.

131. The Actual Card. The actual card which would probably be made by such an engine would be represented by the dotted line which falls within the theoretical card, and is shown separately in the card drawn in Fig. 30. On this card, b is known as the **point of cut-off,** the line ab is known as the **steam line,** the line bc is known as the **expansion line,** c is known as the **point of release,** the line de is known as the **back-pressure line,** e is known as the **point of compression,** ef is known as the **compression line,** f is known as the **point of admission,** and fc is known as the

admission line. The steam line falls below the theoretical line, since a difference in pressure is necessary to force the steam through the valve ports at a velocity which usually ranges from 5000 to 10,000 feet per minute and upward. In order to clear the cylinder promptly, it is necessary to have the point of release come before the end of the stroke, as otherwise too much work would be required during the back stroke to force the steam out of the cylinder. The back-pressure line will be above the theoretical back-pressure line, on account of the difference in pressure necessary to force the steam through the exhaust ports at a velocity ranging from 3000 to 7000 feet per minute. Since the valves open and close gradually and not instantly, the card will have rounded corners at the points of cut-off, release and compression.

132. Power of the Engine. As in the case of any other pressure-volume diagram the ordinates of the indicator card are proportional to the pressure of the working fluid, the abscissæ are proportional to the change in volume, and the area is proportional to the work done by the working fluid. It is usual to compute the power of the engine from the area of its indicator card. By dividing the area of the card by its length (both in inches) we will obtain the height of the mean ordinate (also in inches). The height of this mean ordinate multiplied by the "scale of the spring" will be equal to the average absolute pressure in pounds per square inch on the piston throughout the working stroke minus the average pressure during the back stroke, a quantity which is termed the **mean effective pressure.** If we multiply the mean effective pressure by the net area of the piston and that by the length of stroke, we will have the work done in foot-pounds per stroke. Multiplying by the number of strokes per minute, and dividing by 33,000, we obtain the indicated horse power of the engine, or the rate at which heat energy is transformed into work in the cylinder by the action of the working fluid. In the case of a double-acting engine, it is necessary, when the mean effective pressure and the net piston area of the two sides of the piston are materially different, to calculate the power for each end of the cylinder separately, and to add to the results.

133. Methods of Governing. In order that a steam engine shall be a useful source of power, it is necessary that its speed shall be very nearly constant. This means that the power generated by the expanding steam within the cylinder shall be equal to the friction losses in the engine plus the power required by the machinery which the engine drives. In order to secure this continuous adjustment of the power developed within the cylinder to the varying needs of external machinery, it is necessary to control the quantity of steam admitted to the cylinder during each stroke. We may accomplish this in two ways. We may, by causing the steam to pass through a restricted opening, reduce the

pressure at which a given volume of steam is admitted to the cylinder. Or we may, by closing the inlet valve earlier in the stroke, reduce the **volume** of steam admitted at the cylinder at boiler pressure. The first method of controlling the speed of the engine is known as **throttle governing.** The second method of controlling the speed of the engine is known as **cut-off governing.** The effect of throttle governing upon the indicator card given by a steam engine is shown in Fig. 31. The heavy outline shows the form of the card given by the engine when the throttle valve, which controls the flow of steam to the cylinder, is wide open. The light outlines show the theoretical form of the steam and expansion lines of the card when the throttle valve is closed to a greater or less degree. In the case of an actual engine, the cards would not be of the form shown in Fig. 31, but rather of the form shown in Fig. 32. Since the speed of the piston is greatest at the middle of the stroke, the effect of the throttle valve in reducing the pressure of the steam is greatest at that point; consequently, the steam lines have in practice the form shown in Fig. 32, rather than that shown in Fig. 31.

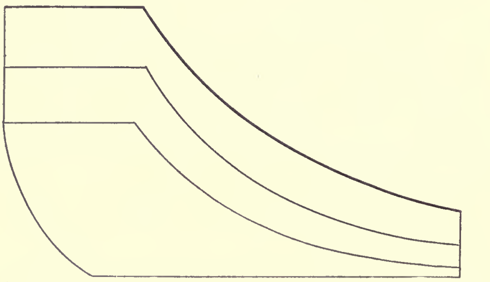


FIG. 31.—Theoretical cards from a throttle governed engine.

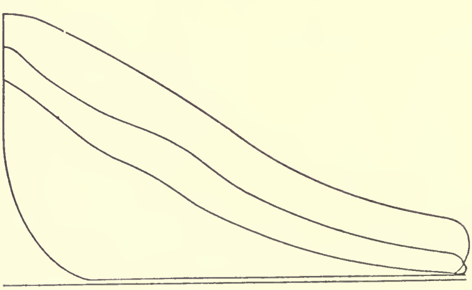


FIG. 32.—Actual cards from a throttle governed engine.

134. The Throttling Governor. The construction and operation of a throttling governor may be understood by reference to Fig. 33. In the figure, *a* is a steam pipe supplying steam to the engine to be governed and *B* is a balanced valve in this pipe. When the valve is at its highest position, the pipe is wide open and the steam can flow freely from pipe *a* into the steam chest *C* by the route shown by the arrow. When, however, the valve is forced down, the area of the opening through which the steam may flow is very greatly diminished, so that a large difference in pressure is required to cause the quantity of steam taken by the engine to flow through the restricted opening. Since this difference in pressure is used

in forcing the steam through the throttle valve, it is not available to drive the piston of the engine, and the power of the engine is therefore reduced. The valve *B* is attached to the rod or stem *d*, which protrudes through the stuffing-box *e*. This rod is forced upward by the helical spring *f*. Two weights or balls, *gg*, are pivoted to the arms *hh*, which are caused to revolve by the gearing shown. The centrifugal force so developed causes the weights to fly outward, and to draw down the valve stem against the resistance of the spring. As the engine speeds up, centrifugal force throws the balls still further out, drawing down the stem, and closing the throttle valve. When the engine slows down, the spring forces the stem up, opening the throttle valve. Any increase above the normal speed will thus reduce the steam supply to the engine, while any decrease below the normal speed will increase this steam supply. The governor is operated by means of a belt or other form of gearing connecting it with the shaft of the engine.

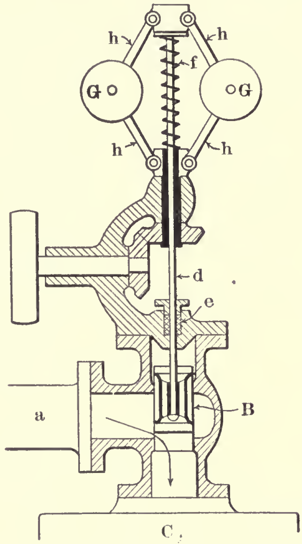


Fig. 33.

135. Cut-off Governing. The variation in the form of card given by an engine governed by means of a variable cut-off governor is shown in Fig. 34. By shortening the cut-off, the quantity of steam admitted, the area of the card, and the power of the engine are all reduced. The heavy outline shows the form of card given at normal load. The light outlines show the form of card at light load, while the dotted line shows the forms of card when the load reaches the maximum.

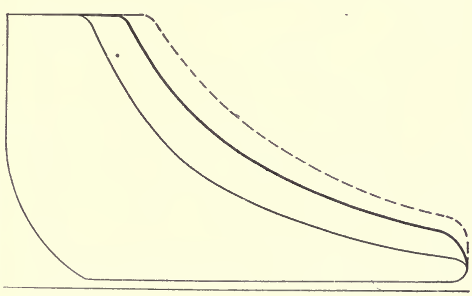


Fig. 34.—Actual cards from a cut-off governed engine.

136. Comparison of Methods of Governing. An inspection of Fig. 35 will show that cut-off governing is preferable to throttle governing, from the standpoint of steam economy. In the figure the heavy outlines show the card given by both a throttle-governed engine, and a cut-off governed engine, at full load. Both engines take the same quantity

of steam, and perform the same quantity of work, and are therefore equally efficient. The light outlines bound the cards given by the same engines at a lower load. The engines, as before, take the same quantity of steam. However, the cut-off governed engine does the work represented by the area *abcde,* while the throttle-governed engine does the work represented by the area *fgcde.* The latter is less than the former by the shaded area, and the efficiency of the throttle-governed engine is correspondingly less than that of the cut-off governed engine. Consequently, we find that practically all first-class modern engines are governed by a variable cut off, and not by throttling.

The mechanism used for producing a variable cut-off is usually much more costly and complicated than that used in throttle governing. The general types of valves used for this purpose and the mechanisms employed for moving them will be discussed in the remaining paragraphs of the present chapter, in connection with the descriptions of different types of engines now in use.

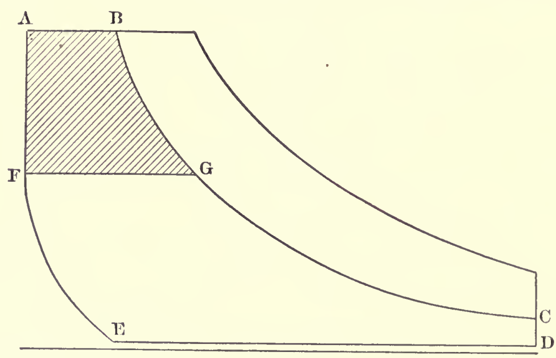


FIG. 35.—Comparison of throttle and cut-off governing.

137. The Common Slide Valve. The form of valve most commonly employed in the smaller sizes of steam engines is known as the **slide valve.** The simplest form of the slide valve is that shown in section in Fig. 36, and known as the **D valve** on account of its shape. In this figure, which is a longitudinal section through the cylinder and steam chest, the valve is the blackened part marked V. The valve covers the ports P and P' leading to the head and crank end of the cylinder. Steam enters the valve chest from the steam pipe, and as the valve moves to the right into the position shown, the port P is uncovered, allowing the steam to flow into the cylinder and propel the piston to the right. At the same time the steam contained in the crank end of the cylinder escapes through the port P, into the exhaust port E, by the route shown by the arrow. The valve is caused to move back and forth by the action of an **eccentric**

on the engine shaft. The **eccentric** is a disk whose diameter is much greater than that of the shaft to which it is attached, and which acts in the same manner as a crank, since its center does not coincide with the center of the shaft. As the piston continues to move to the right, the valve begins to move to the left, finally shutting off the supply of steam, and expansion begins. When the piston approaches the end of its stroke, the valve still moving to the left shuts off the port P' from the

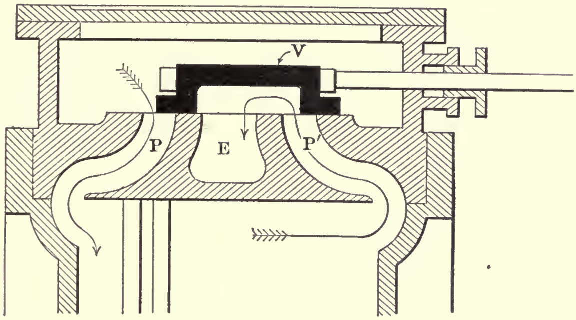


FIG. 36.—The D valve and ports.

exhaust port, and compression begins in the crank end of the cylinder. When the piston reaches the end of its stroke, the valve will have moved to the left sufficiently to uncover the port P' to the steam in the valve chest, and the port P to the exhaust port. During the back stroke of the engine, the same series of events occurs, except that the opposite end of the cylinder is involved in each case.

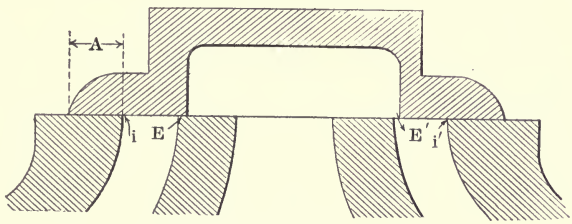


FIG. 37.—Steam and exhaust lap.

In order to accomplish these results, it will be seen that it is necessary when the valve is in its central position, for the two ends of the valve to extend some distance beyond the **inlet edges** (i and i') of the ports, as is shown in Fig. 37. The distance A is known as the **steam lap** of the valve, and may be the same or may be different for each end of the cylinder. It is usual for the **exhaust edges** of the ports to coincide with the **exhaust edges** of the valve, when the valve is in the central position. Sometimes the ports are slightly uncovered as at E', and sometimes they are cov-

ered as at E, with the valve in this position. The distance from the exhaust edge of the port to the exhaust edge of the valve is termed **negative** exhaust lap in the first case, and **positive** exhaust lap in the second. The distance which the valve moves in going from one of its extreme positions to the other is called the **travel** of the valve, and is twice the **eccentricity** or **throw** of the eccentric.

By changing the travel of the valve (i.e., the eccentricity of the eccentric) and the angular distance between the eccentric and the crank, it is possible to change the point in the stroke at which cut-off occurs. Such a change will also affect the point in the stroke at which admission, exhaust, and compression begins, but these changes will not be as great as the change in the time of cut-off. If the eccentric be connected to a governing mechanism which will automatically adjust its eccentricity and position in accordance with the power required of the engine, we may effect cut-off governing by the slide valve. An engine so governed is usually termed an automatic engine.

138. Defects of the Common Slide Valve. The D valve has the following defects: First, on account of the rather long and crooked steam passages, the clearance volume of the engine will be excessive. The result, as will appear in the next chapter is a considerable waste of steam. Second, the area of the surfaces enclosing the clearance space will be very large, which will be shown in Chapter X to result in a great waste of steam. Third, the valve opens and closes gradually instead of promptly, and the card given by the engine shows to an excessive degree the effects of wire drawing. As a result, the power given by the engine for a given weight of steam is considerably less than if the valves opened and closed promptly. Fourth, the pressure of the steam on the back of the valve forces it down upon its seat, thus causing it to wear excessively and to require a considerable amount of power to drive it. Fifth, when the plain slide valve is used in an automatic engine, on account of the shifting of the points of compression and release, the card is greatly distorted from its proper form, except at some particular load, which reduces the power obtained from the engine for a given steam consumption. In order to overcome these several disadvantages, a great many types of valves have been invented.

The excessive clearance and area of clearance surfaces may be reduced by lengthening the valve, so that the ports may be made short and direct. By doing so, however, we greatly increase the total pressure of the steam upon the valve, and therefore the friction and wear. In order to reduce this source of loss, the valve is usually, in the better class of engines, made of such a form that the pressure of the steam is balanced. Such a valve is termed a **balanced valve.** In order that the valve shall open and close promptly, it is usual in all but the smallest engines to use what

is termed a double-ported valve; that is one which will admit the steam at two edges instead of at one edge only. In order to avoid the shifting of the points of release and compression, as the load upon an automatic engine varies, an auxiliary valve, whose purpose it is to regulate the point of cut-off, is sometimes employed. Such a valve is called a **riding cut-off valve.**

139. Balanced Slide Valves. The simplest method of balancing the slide valve is to use a valve which is cylindrical in form, instead of a flat valve. Such a valve is shown in Fig. 38. The steam is admitted at one end of the valve chest, and since the valve is hollow, it can pass from one end to the other. The action of the valve may be readily inferred from the drawing, and it differs in no way from the action of the D valve.

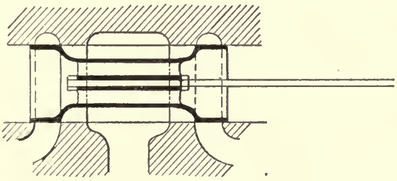


Fig. 38.

A second method of balancing the slide valve is that shown in Fig. 39, in which the valve is a perfectly flat rectangular plate covered by a second plate having in it slight recesses exactly opposite to the steam and exhaust ports, and connected to them by passages through the valve itself. Since the pressure on both sides of this valve is exactly the same, the friction resisting its motion is negligible. The objection to both these methods of balancing slide valves is that the valves leak, about 25 per cent of the

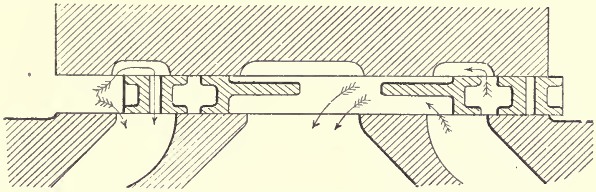


Fig. 39.—Straight-Line valve.

steam consumption of engines using these types of valves being due to such leakage. Various other methods of balancing slide valves are in use, and the reader is referred to Halsey's "Slide Valve Gears" for a more complete treatment of the subject.

140. Multi-ported Slide Valves. The simplest form of the double-ported valve is the Allen valve illustrated in Fig. 40. The port cored in the body of the valve, permits the steam to pass into the steam ports by the two routes indicated by the arrows. A displacement of the valve of any given small amount to the right of the position shown will

increase the width of the port opening by twice the amount of the displacement.

A method of double-porting a balanced slide valve is illustrated in Fig. 39, which represents the Straight-Line valve as applied to the Straight-Line engine. The path of the steam through the double ports may be inferred from the arrows, which show that the steam is admitted at two edges at the same time. Double porting the slide valve serves

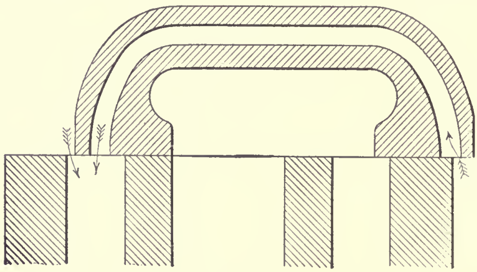


Fig. 40.—Allen double ported valve.

to reduce the friction of the valve for a given port opening, and therefore the friction and wear. The same rapidity of port opening may of course be obtained by using a common *D* valve with twice the travel, but it is not always advisable to give a valve such an amount of travel.

141. Riding Cut-offs. The simplest form of the riding cut-off valve is illustrated in Fig. 41, and is known as the Meyer valve. The main

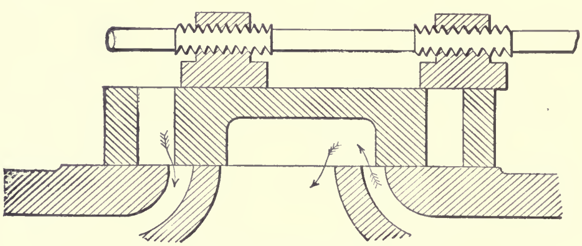


Fig. 41.—Meyer riding cut-off valve.

valve acts as a seat upon which the cut-off valve slides, the cut-off valve being driven by a separate eccentric. The cut-off valve is formed in two parts, as shown, in order that the point of cut-off may be altered by altering the distance between them. This is accomplished by threading the two halves of the valve to the valve stem with a right- and left-handed thread in order that by turning the valve stem, they may be made to approach or recede. This type of valve is much used in slow-moving engines operated without governors, such as air compressors and pumping engines. It is not, however, as satisfactory

as other forms of riding cut-off, such for instance as the Buckeye valve, which may be found described in Halsey's book, to which reference has already been made.

142. Application of Slide-valve Engines. The slide-valve engine is usually built in sizes up to 300 or 400 horse-power for stationary service, and in larger powers when compact engines of high speed of rotation are desired. It is not as economical in the use of steam as are other types of engines, and is being steadily displaced by the steam turbine and by the four-valve engine. In stationary service, in the smallest sizes, a fixed eccentric and throttle governor are commonly used. In the larger sizes, the valve is made double-ported and balanced and an automatic shaft governor is used. In spite of the greater economy which it offers the riding cut-off is seldom employed.

The slide-valve engine finds its principal application in high powers in connection with locomotive and the marine engines. The valve employed for high steam pressures is usually a double-ported piston valve, while for low pressures a double-ported balanced flat valve is usual. Neither locomotives nor marine engines are provided with governors, and the eccentrics are fixed. However, the point of cut-off and the power of the engine, in both cases, is controlled by means of an arrangement termed a **link motion.** The Stevenson link motion was formerly employed almost exclusively, but other forms, notably the Waelchert valve gear, have been found more suitable for very large powers and high steam pressures. These valve gears are manually controlled, the engineer determining the speed and power of the engine by the position of the link motion. Although slide valves have been used exclusively in locomotive and marine engine service, there is just as good reason to believe that the employment of other types of valve motion would give superior results in this class of service as in any other. It is reasonable to suppose, therefore, that four-valve engines will be eventually employed in locomotive and marine service, since they are giving most excellent results in stationary service.

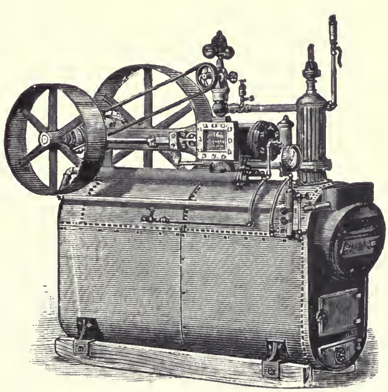


Fig. 42.—Portable engine and boiler. Engine with throttling governor.

In Fig. 42 may be seen an illustration of a plain slide-valve engine with a throttle governor mounted upon a portable boiler. This is a type of power plant often employed in agricultural service. In Fig. 43

is an illustration of a single-cylinder high-speed automatic engine with a double-ported balanced valve, such as is usually employed in stationary service when the power required is small, and the engine runs non-condensing. This engine is equipped with a shaft governor. Fig. 44 is an illustration of the cylinder and drive wheels of a locomotive equipped with a Waelschert valve gear.

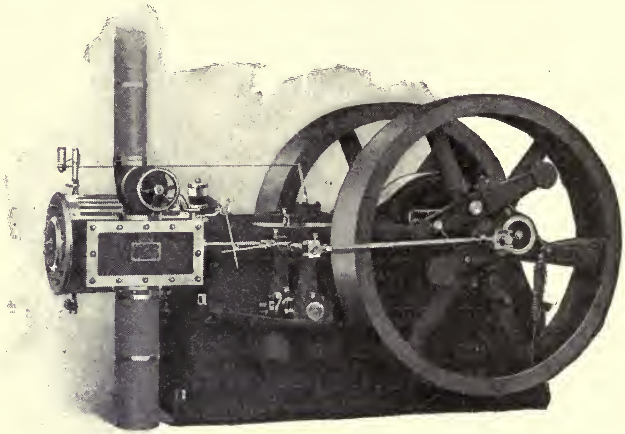


Fig. 43.—High speed simple engine with "automatic" cut-off governor.

A locomotive of this type has two cylinders of largesize, uses steam of very high pressure, and operates at very high speed, developing from 700 to 1200 horse-power in each cylinder. On account of the large number of locomotives in service, and their very great aggregate power, the matter of locomotive engine efficiency is of immense practical

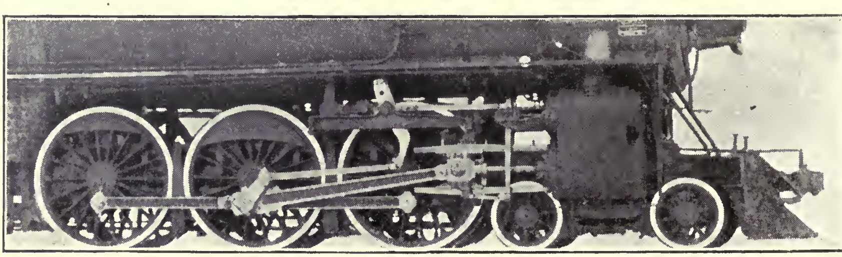


Fig. 44.—Cylinder and drive wheels of a locomotive with Waelschert gear.

importance. In Fig. 45 may be seen an illustration of a triple-expansion, four-cylinder marine engine, such as is commonly built for naval service. These engines are of immense power for their size and weight and run at very high speed. The high-pressure cylinders are equipped with piston valves and the low-pressure cylinders with balanced flat

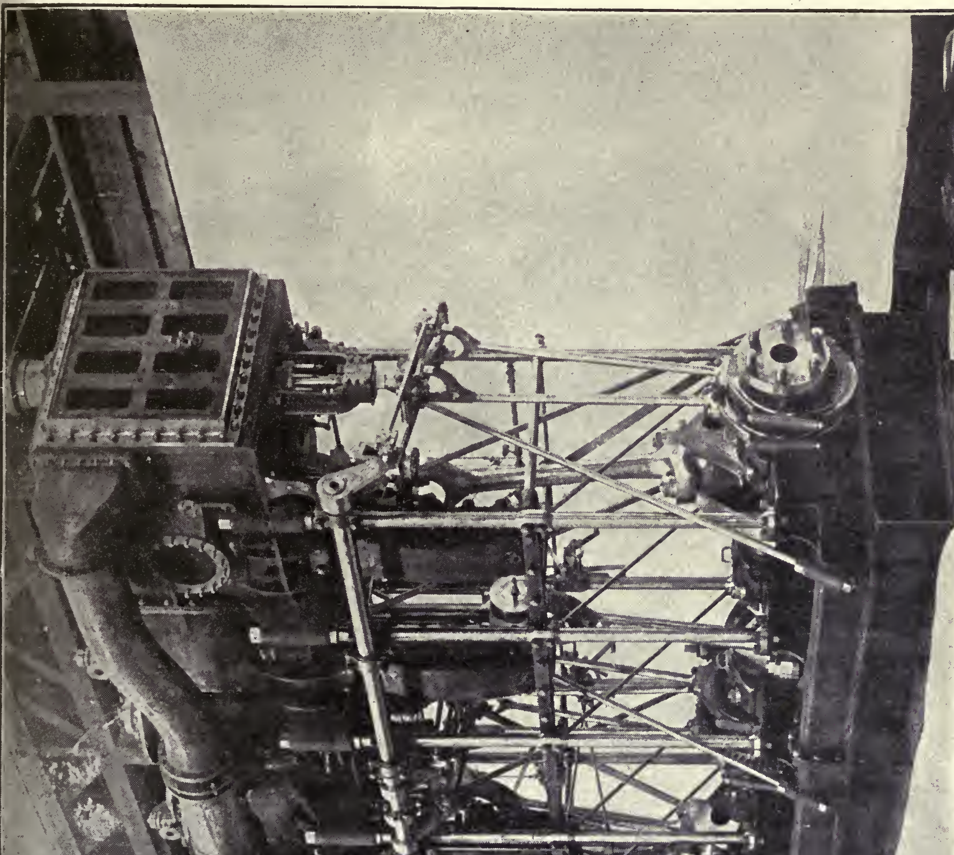


FIG. 45.—Four-cylinder triple-expansion naval engine built by the Fore River Shipbuilding Co.

valves, all operated by Stevenson link motion. This link motion is not moved directly by the engineer, as is the case with the locomotive, but is operated by means of a steam cylinder, which is controlled by the engineer, the links being too large and heavy to be manually operated.

The variety of slide valves in actual use is so numerous and the methods of designing them are such as to forbid an adequate treatment of the subject in a work of this kind. For a full treatment of American slide valves and the best methods of designing them the reader is referred to Halsey's " Treatise on Slide Valve Gears."

143. The Corliss Engine. The Corliss engine makes use of valves of the form shown in Fig. 26, in order to avoid the disadvantage accompanying the use of slide valves. By the use of the Corliss valve the ports may be made short and direct. The clearance volume and clearance area are reduced to a minimum, which results in greatly increasing the steam economy. In addition the mechanism which operates the valves is so arranged that the valves are caused to open and close promptly, thus avoiding a loss of power from wire drawing. The closing of the inlet valve at the point of cut-off is effected in such a way as to leave the points of admission, release, and compression unchanged, which permits them always to occur at the proper point in the stroke.

144. The Corliss Valve Motion. The mechanism employed for operating the valves of a Corliss engine is illustrated in Fig. 46. W is the **wrist plate,** which turns upon a pin fastened to the side of the cylinder. It is operated by the **reach rod** R, which is pinned at the other end to the rocker arm A, which in turn is operated by the **eccentric rod,** the other end of which is fastened to the **eccentric strap,** which embraces the **eccentric** E. Attached to the wrist plate are four rods termed **valve rods** leading to the levers which operate the four valves of the engine. The rods B and B' are pinned to the arms C and C', which in turn are keyed to short shafts termed **valve stems** which serve to rotate the exhaust valves. As the wrist plate is vibrated by the action of the eccentric, the valves are caused to move, opening and closing at the proper time. The action of the wrist plate is such that at the end of their travel, while they are closed, the valves are practically stationary, as will be the case with the left end or head end exhaust valve, when the wrist plate is in the position shown in Fig. 47. This allows of a rapid opening of the valve without an excessive valve travel.

The valve rod D is pinned to the bell crank, or **latch arm** F in Fig. 48, which turns freely upon the inlet-valve stem. Pinned to the latch arm is a latch which when the arm is in the position shown, catches a block affixed to the **inlet-valve arm** G, and as the latch arm is drawn to the right the valve arm is caused to rotate, opening the valve. When this rotation has proceeded for a sufficient length of time to allow the

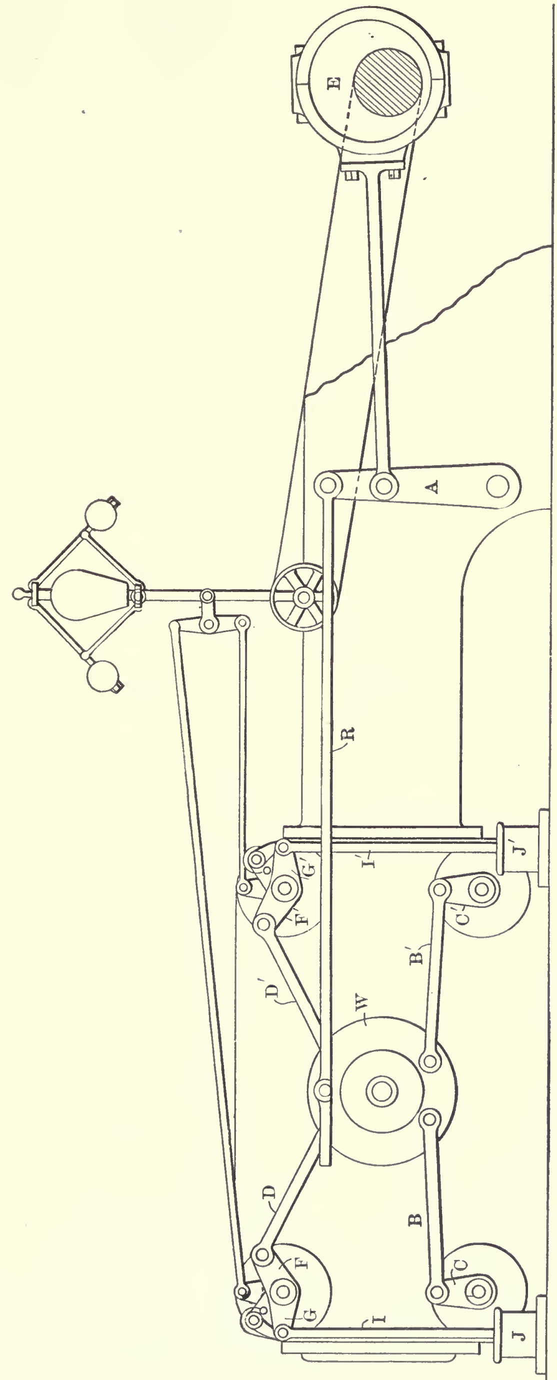


FIG. 46.—Corliss valve motion. Wrist plate on center.

desired cut-off, the latch strikes the cam H (whose position is fixed by the governor) which pushes up the latch and releases the block. The rod I, pinned to the valve arm, leads to the dashpot J. Within this dashpot is a piston, the raising of which creates a suction which returns the valve

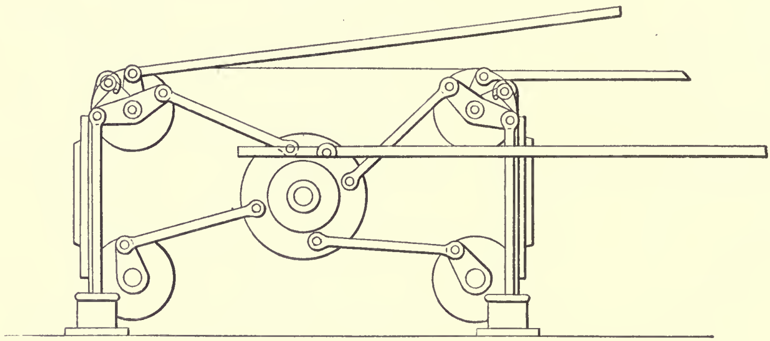


FIG. 47.—Corliss valve motion with wrist plate in extreme position.

to the closed position the instant the cam causes the latch to release the block. This piston is so arranged that just before it strikes the bottom of the cylinder it compresses a quantity of air, which prevents the violent blow or jar which would otherwise result. By this device, a

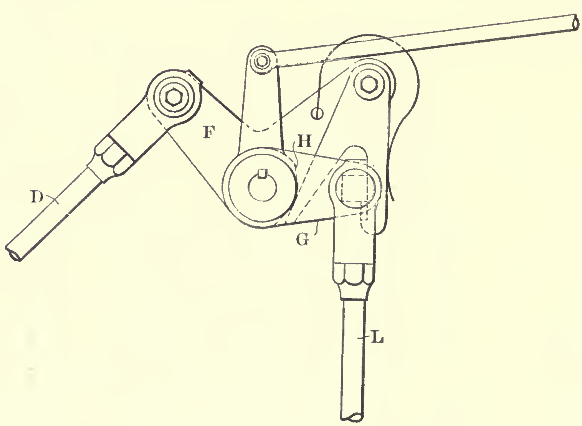


FIG. 48.—Inlet valve gear.

very rapid closing of the ports is secured. A still more rapid opening and closing of the ports of a Corliss engine may be secured by the use of double-ported valves, such as are shown in section in Fig. 49.

It is often desirable to use two eccentrics, operating two wrist plates,

one of which moves the exhaust valves and the other the inlet valves. This permits of a much later cut-off than is possible when one eccentric is used. When Corliss engines are required for railway or other service where the variation in load is extreme, this type of engine is preferred. When the load is fairly constant, the single eccentric type is equally as satisfactory and economical.

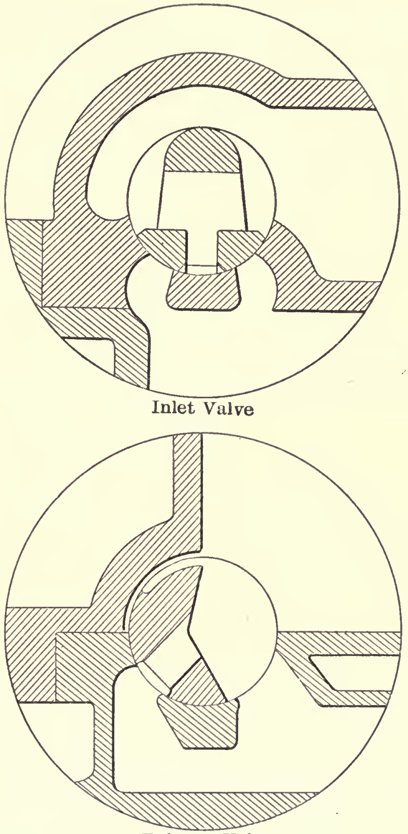


FIG. 49.—Sections through Corliss inlet and exhaust valves.

145. The Four-valve Engine. A type of engine known as the four-valve automatic engine is rapidly coming into favor for smaller powers (i.e., up to 300 to 500 horse-power). The exhaust valves of this type of engine are operated in the same way as are the exhaust valves of a Corliss engine. The inlet valves, however, are operated directly from the inlet-valve wrist plate, the inlet-valve rods being pinned to the valve arms, which are keyed to the valve stems. The cut-off is effected by changing the throw and angular position of the eccentric which operates the inlet-valve wrist plate. All of the advantages of the Corliss engine are realized in the four-valve automatic engine, excepting the prompt closing of the inlet valve at cut-off. The form of card given by the four-valve engine is shown in Fig. 50. There is no material difference between the steam economy of the four-valve automatic engine and of the Corliss engine, but the four-valve automatic, since it may be operated at higher speeds, is the cheaper type, and since it

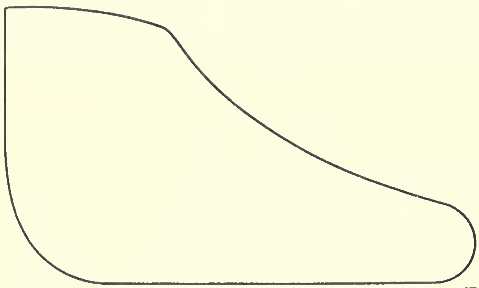


FIG. 50.—Typical card from a four valve engine.

is simpler, it is less likely to get out of order. The governor employed with the four-valve automatic engine is a shaft governor, while that employed with the Corliss engine is usually a fly-ball governor, such as is shown in Fig. 46.

146. Gridiron Valves. Another type of valve which is used in a good many makes of steam engines is the gridiron valve, which is illustrated in

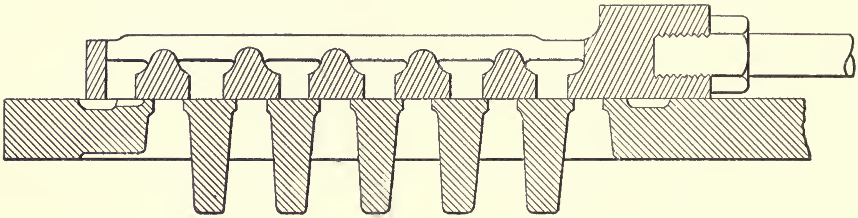


FIG. 51.—Section through a gridiron valve.

Fig. 51. The seat upon which this valve rests contains a series of parallel slots separated by metal bridges somewhat wider than the slots, while the valve itself contains a similar set of slots. When the slots are opposite one another, the steam passes through the valve. When, however, the

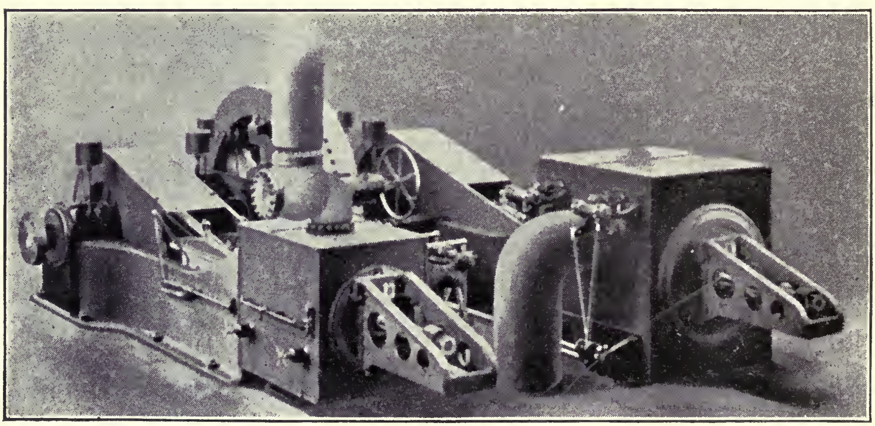


FIG. 52.—McIntosh-Seymour cross compound engine.

bridges of the valve cover the slots in the seat, as shown in the drawing, steam cannot pass through. Various types of mechanism are employed for operating gridiron valves. The McIntosh-Seymour engine, illustrated in Fig. 52, is fitted with gridiron valves, and is an example of a type of mechanism often employed for operating them. The engine equipped with gridiron valves is equally as economical as the Corliss engine, and was formerly equally as cheap to build. Changes in shop methods have

resulted in making the Corliss type the cheaper one to build, so that most large engines are now equipped with Corliss valves.

147. Poppet-valve Engines. A type of valve very much employed in European practice is known as the poppet valve. The poppet valve is of two forms, the one shown in Fig. 53 being known as a plain poppet valve and that shown in Fig. 54 as a balanced valve. In steam engine work the balanced poppet valve is usually employed. Poppet-valve engines are usually four-valve engines, although poppet valves are sometimes employed in pairs and sometimes in connection with Corliss or other types of valves. Balanced poppet valves have the disadvantages of requiring a large clearance volume and of exposing a considerable clearance area to the action of the steam. They are, however, better adapted to the use of superheated steam, and are tighter than are other forms of valves, and are therefore much used in connection with superheated steam. A tripple-expansion pumping engine in which poppet valves are used for the exhaust valves of the intermediate cylinder, and the inlet and exhaust valves of the low-pressure cylinder, is illustrated in Fig. 55. In a poppet-valve engine, as in the Corliss engine, cut-off is effected by releasing the valve from the control of the opening mechanism, and allowing it to close quickly by the action of a dashpot.

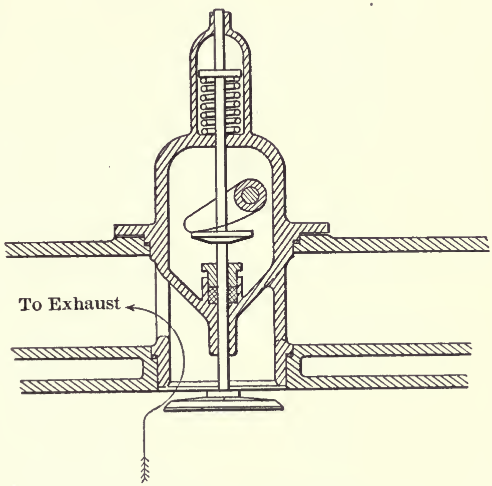


FIG. 53.—Plain poppet exhaust valve.

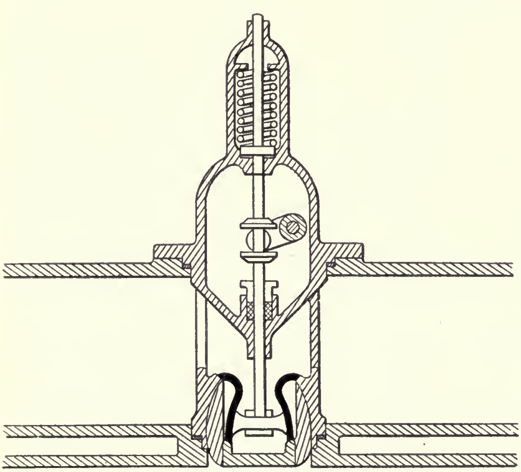


FIG. 54.—Balanced or double-beat poppet inlet valve.

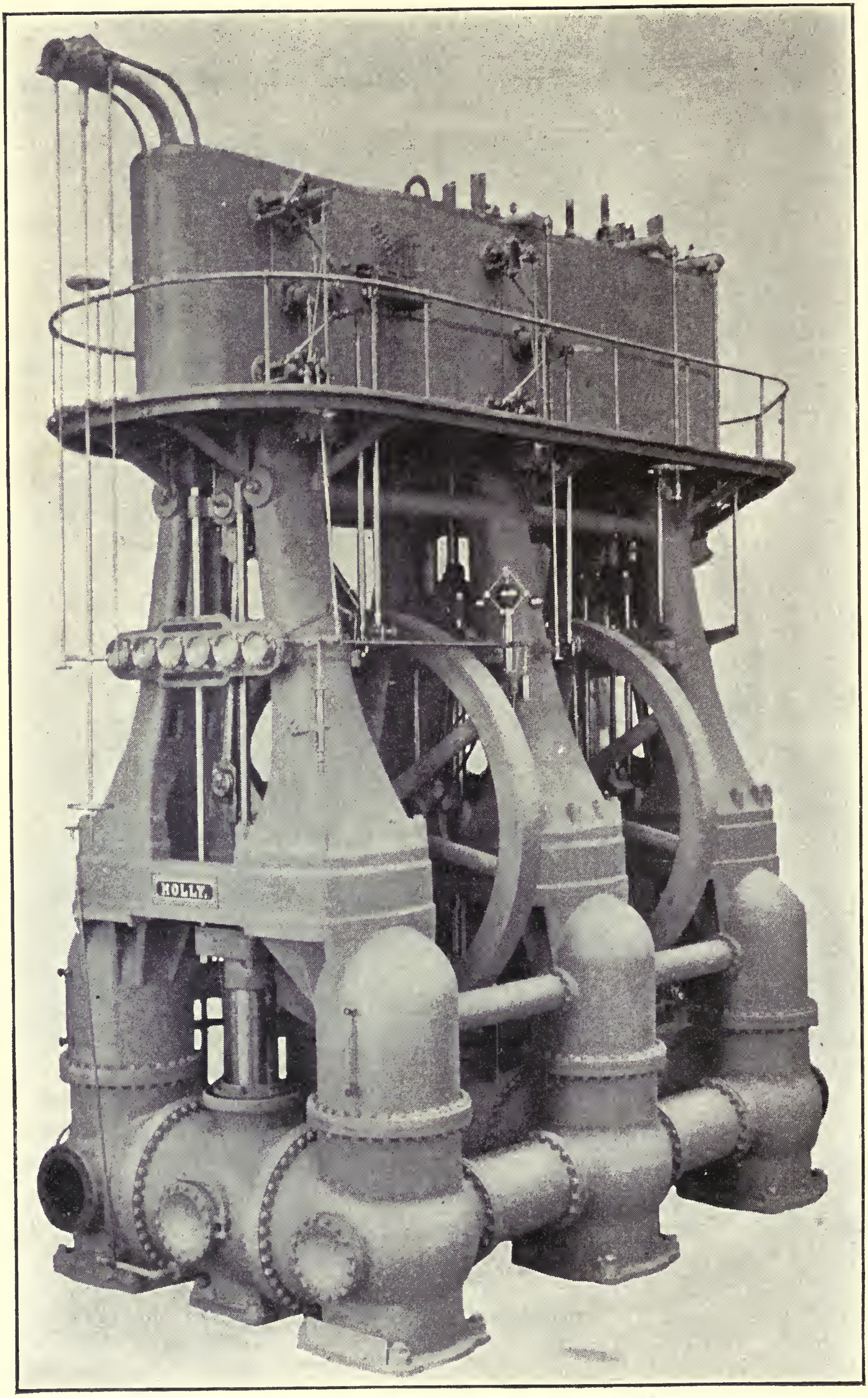


FIG. 55.—Vertical triple-expansion pumping engine.

Excellent results in steam economy have been obtained from poppet-valve engines, but these results are to be credited rather to the fact that highly superheated steam was employed than to any excellence inherent in the type of valve. The use of plain poppet valves in the low-pressure cylinder of a steam engine permits of greatly reducing the clearance volume and the resulting loss. By employing this type of valve, engines have been designed in which the clearance volume was only 0.35 per cent of the swept volume. In such a case, the clearance loss is exceedingly small.

148. Rotary Engines. Many attempts have been made to so design the steam engine as to avoid the use of a reciprocating piston and cross-head, applying the expansive pressure of the steam directly to rotating parts. Engines operating on this principle are usually called rotary steam engines. The rotary steam engine has not proved successful, for two reasons. First, the form of cycle which must be employed with any of the possible mechanisms is wasteful of steam, and secondly, the friction and wear of the parts are excessive. Since the mechanical efficiency of a well-designed reciprocating engine is high, there is no practical reason for the use of a rotary engine except a possible reduction in the volume and weight of the engine. This advantage, however, is outweighed in all cases by the larger steam consumption and more rapid deterioration of engines of this type.

149. Special Forms of the Steam Engine. Many special forms of the steam engine are employed for special service. The steam hammer, for instance, is a special form of an engine with manually operated valves. Direct-acting steam pumps, the locomotive air pump, the steam steering engine, the pulsometer, the steam drill and other types are examples of highly specialized types of steam engines, adapted to work under peculiar conditions, or to perform unusual kinds of service. Such engines are usually of peculiar construction mechanically, and are almost invariably very wasteful in their use of steam, and are employed only because they offer superior advantages in the matter of cheapness, simplicity, adaptability, or ruggedness of mechanism.

PROBLEMS

1. An engine takes steam of 98 per cent quality at a pressure of 100 lbs. absolute and rejects steam at a pressure of 15 lbs. absolute. The engine used 30 lbs. of steam per horse-power per hour. Find the efficiency. Ans. 8.6%.

2. Find the loss due to radiation in the above problem, if the steam exhausted is of 91 per cent quality. Ans. 20 B.T.U. per pound of cylinder feed.

3. A pressure of 75 lbs. gage raises the pencil point of an indicator $1\frac{1}{2}$ ins. above the atmospheric line. What is the scale of the spring? Ans. 50 lbs.

4. An indicator card has an area of 3.5 sq.ins. and a length of 3 ins.; find the mean effective pressure when the scale of the spring is 40 lbs.

Ans. 46.7 lbs. per square inch.

5. The area of the piston of an engine is 100 sq.ins. The mean effective pressure is 40 lbs. per square inch. The length of stroke is 2 ft. and the engine makes 150 revolutions per minute. The engine is double acting. Compute its horse-power.

Ans. 72.7 H.P.

CHAPTER IX

STEAM CYCLES

150. The Carnot Cycle for Dry and Saturated Steam. The principal factor in the cost of steam engine operation is the efficiency of the engine, which may be defined as the ratio of the work done by the engine to the heat supplied to the engine. The efficiency of the engine depends primarily upon the efficiency of the cycle performed by the working fluid in the engine cylinder. It is therefore in order to investigate the efficiencies of the various cycles employed in actual engines, and the methods by which these efficiencies may be increased. This chapter will not deal with those losses which are due to the imperfection of the materials or mechanism of the engine, but only with those which are due to the cycle performed by the working fluid.

In theory there are many different cycles which may be performed by the working fluid of a steam engine. The most efficient of these is the Carnot cycle. In order to carry out the Carnot cycle in a steam engine using dry and saturated steam, the steam must be evaporated in the cylinder instead of in a separate boiler, and condensed in the cylinder, instead of being rejected to the air or to a separate condenser. The indicator card of the Carnot cycle for a steam engine is shown in Fig. 56. The volume of the water is the length of the abscissa to point *a*. The volume of the steam formed is the length of the abscissa to point *b*. As soon as the water is completely evaporated, the steam being dry and saturated, adiabatic expansion begins, continuing to point *c*. During this adiabatic expansion some of the steam condenses, as was shown in Art. 109, Chapter VI. When the steam has expanded to the temperature of the cooler, the back stroke commences, and the steam is compressed and condensed at constant pressure by the action of the cooler, until point *d* of the diagram is reached. At this point the cooling ceases, and the mixture of steam and water is then compressed adiabatically, which increases the temperature of the mixture, and since the mixture is very wet, condenses the remaining steam. At the end of this compression all of the

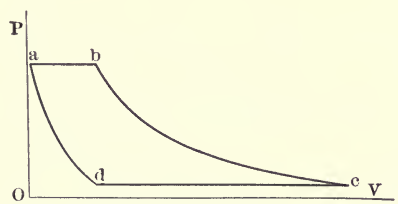


Fig. 56.—Carnot cycle for dry and saturated steam.

steam has been condensed, and the water has the temperature of vaporization corresponding to the pressure at a.

The efficiency of this cycle depends solely upon the absolute temperature during the isothermal expansion and compression from a to b from c to d, and was shown in Chapter IV to be

$$E=\frac{T_1-T_2}{T_1},$$

in which E is the efficiency of the cycle, T_1 is the absolute temperature of the steam during evaporation, and T_2 is the absolute temperature of the steam during condensation. Such a cycle is obviously impracticable in the case of steam, as no mechanism can be devised which will reproduce it exactly. We may, however, reproduce it approximately by methods which will be described later.

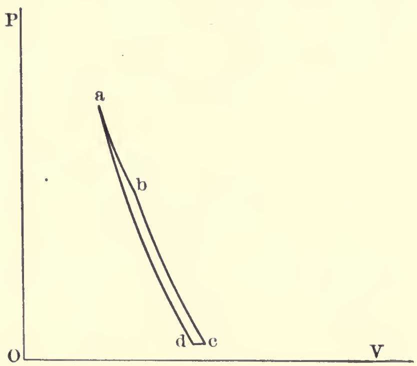


Fig. 57.—Steam initially superheated.

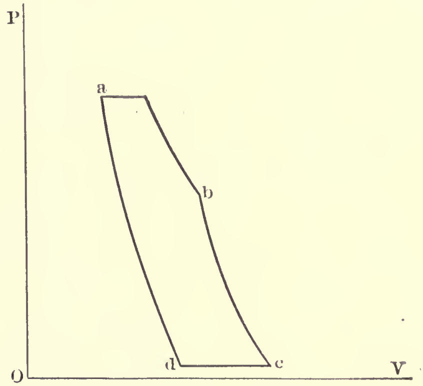


Fig. 58.—Steam initially wet.

151. The Carnot Cycle for Wet or Superheated Steam. The Carnot cycle may be performed when using wet or superheated steam as a working fluid. If the steam is wet at point b, the card will be similar in form to that shown in Fig. 56, but the volume at b and at c will be less than when the steam is dry at the beginning of expansion. It may be remarked in this connection that it is not necessary that the steam be entirely condensed at point a to perform a Carnot cycle, provided only that it returns to its initial state. If the steam is highly superheated at the beginning of isothermal expansion and the temperature range of the cycle is not too great, the steam will remain superheated throughout the entire cycle and the card given by the engine be almost identical with that which would be given by a perfect gas. The form of the card is the same as that shown in Fig. 14, Chapter IV, for a gas. In case the steam is not highly superheated at the beginning of isothermal expansion it will become wet as a result of the adiabatic expansion, the isothermal compression line will also be isobaric and the form of the card will be that shown in Fig. 57. Only a small portion of the steam may be condensed during isothermal compression, in this case, since the whole mass of steam and water must be returned to its original state by the adiabatic compression. The steam may be initially wet, and become superheated during isothermal expansion, in which case the card will have the form shown in Fig. 58. When

superheated steam is employed as the working fluid in a Carnot engine, the volume of cylinder must be larger for a given weight of working fluid than when saturated steam is used. Also the work performed by a given weight of superheated steam will be very much less for the same range of temperature than would be performed by the same weight of saturated steam. The efficiency of the cycle for a given temperature range, is the same, whether wet, dry or superheated steam is employed. Neither the Carnot cycle itself nor any approximation to it is ever actually employed in connection with superheated steam, on account of the very great cylinder volumes required to obtain very moderate amounts of power.

152. The Rankine[1] Cycle. A second steam cycle is one which is known as the Rankine cycle. Since this is the most efficient cycle which may be performed by steam without evaporating and condensing the working fluid within the engine cylinder, it has been adopted as the standard of efficiency with which the efficiency of all other cycles may be compared. The indicator card of this cycle is shown in Fig. 59. The engine is assumed to have no clearance, and the walls of the cylinder to be non-conductors of heat. Steam is admitted from a boiler from point a to point b, the expansion being isothermal (and isobaric) and the boiler and steam pipe being a part of the expansion chamber. At b cut-off occurs, and the steam contained in the cylinder expands adiabatically to the pressure of the exhaust pipe, as shown by line b–c, some of it condensing during the process. At the end of this expansion the steam is discharged into the exhaust pipe at a constant back pressure, line c–d representing this process of isothermal compression. Line d–a represents the rise in temperature and pressure without change in volume which results when the inlet valve opens.

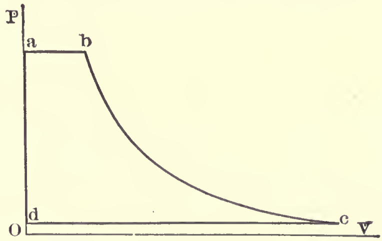


FIG. 59.—Watt diagram of the Rankine cycle.

The efficiency of the Rankine cycle may be found in the following manner: During the period of admission the work done by each pound of steam is equal to the external work of evaporation of steam of the temperature of admission multiplied by the quality of the steam admitted. During the adiabatic expansion the steam loses intrinsic energy, and the work done during expansion is equal to the difference between the intrinsic energy of the steam at the beginning and at the end of the expansion. The work done in forcing the steam out of the cylinder against the back pressure is equal to the external work of evaporation of steam at the temperature of exhaust, multiplied by the quality of the steam exhausted. The work done during the cycle per pound of steam will then be equal

[1] Often termed the Clausius Cycle.

to the sum of the external work of evaporation and the intrinsic energy of the steam admitted, less the sum of the external work of evaporation and the intrinsic energy of the steam exhausted, which is, of course, equal to the difference between the total heat of the steam admitted and the total heat of the steam exhausted. If we know the pressure (or temperature) and quality of the steam admitted, we may compute its total heat and its entropy. The entropy of the steam exhausted is the same as that of the steam admitted, since the expansion is adiabatic. From the known entropy and temperature of the steam exhausted, we may compute its quality and its total heat. The difference between the total heats is the work done per pound of steam admitted. The heat supplied in the boiler to each pound of steam is equal to the total heat of the steam admitted, less the heat of the liquid at the temperature of exhaust. We may therefore express the efficiency of this cycle by the formula

$$E=\frac{H_1-H_2}{H_1-h_2},$$

in which H_1 is the total heat of the steam admitted, H_2 is the total heat of the wet steam discharged, and h_2 is the heat of the liquid at the temperature of exhaust.

It may be shown that the efficiency of the Rankine cycle, like that of the Carnot cycle, is increased by increasing the temperature range of the working fluid. Referring to the formula for the efficiency of the Rankine cycle given in the preceding paragraph, it will be seen that an increase in the initial total heat of the steam will result in an increase of the efficiency of the cycle, since both the numerator and denominator of the fraction will be increased by the same amount, while the total heat of the steam rejected will be diminished, on account of the greater range of expansion. Since the total heat of the steam increases with the pressure, it will be seen that an increase in the initial pressure of the steam must result in an increase in the efficiency of the cycle. An investigation of the properties of steam will show that when it expands adiabatically, between any two temperatures, the decrease in the total heat is greater than the decrease in the heat of the liquid. Consequently, a downward extension of the temperature range of the Rankine cycle will add to the numerator of the fraction expressing the efficiency a larger quantity than it will add to the denominator, and the efficiency of the cycle will be increased by a reduction of the terminal pressure.

In case superheated steam is used in the Rankine cycle, the form of card will be exactly the same as that shown in Fig. 59, except that the form of expansion line will slightly change when the expanding steam reaches the saturation point. The efficiency of the cycle will, of course,

be expressed by the formula already given, but the value of H_1, instead of being the value for the total heat of wet steam, will be the value for the total heat of superheated steam of the given pressure and temperature An investigation of the properties of superheated steam will show that the greater the superheat of the steam, the greater will be the efficiency of the cycle. Since it is practicable to use superheated steam of a much higher temperature than saturated steam, this is a matter of importance in the theory of the economy of the steam engine.

153. The Modified Rankine Cycle. The Rankine cycle described in the preceding paragraph cannot be reproduced in a steam engine, since no engine can be constructed without clearance, or of materials which are perfect non-conductors of heat. However, it would be possible in a non-conducting cyclinder to produce a cycle which is the thermodynamic equivalent of this cycle by the method shown in Fig. 60, which is the theoretical indicator card from an engine operating on a modified Rankine cycle. The engine has the clearance volume represented by the distance from OP to point a. Cut-off occurs at b, adiabatic expansion occurs from b to c to the pressure of the exhaust, the exhaust is discharged at this pressure from c to d, and at d compression begins. The volume at point d is so chosen that by adiabatic compression of the entrapped steam, it will be raised to its initial pressure, temperature and quality, in passing from state d to state a. An engine operating on this cycle has exactly the same efficiency as an engine operating on the Rankine cycle, since the cushion steam does the same work during expansion as is done upon it during compression. However, the volume of its cylinder must be somewhat larger than that of an engine operating on the Rankine cycle, since the volume from c to d in each diagram must be the same in order for the two engines to develop the same power per stroke.

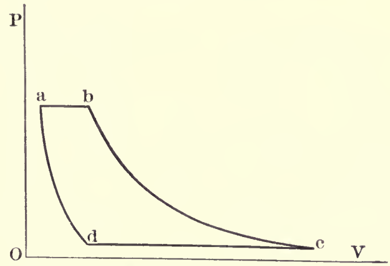


FIG. 60.—Modified Rankine cycle.

154. Computation of a Rankine Cycle. The following example will serve to make clear the method of computing the efficiency of the Rankine cycle for a given range of temperature and pressure. Assume that one pound of steam of a pressure of 150 pounds absolute and a quality of 90 per cent performs a Rankine cycle, being exhausted at a pressure of 2 pounds absolute. The total heat of the steam will be $h+qL=330.2+90\times 863.2=1107.1=H_1$. The entropy of the entering steam will be $n+\frac{qL}{T}=$

$.5142+.90\times 1.0550=1.4637=N_1$. The entropy of the exhaust steam will be the same as that of the entering steam and the entropy of evaporation will be $N_2-n_2=1.4637-0.1749=1.2888$. The quality of the steam exhausted will be

$$\frac{1.2888}{1.7431}=74.0 \text{ per cent.}$$

The total heat of the steam exhausted will be $94.0+.74\times 1021.0=850=H_2$. The heat of the liquid, h_2 is from the tables 94.0 B.T.U. The efficiency of the cycle is therefore

$$E=\frac{1107.1-850.0}{1107.1-\ \ 94.0}=25.0 \text{ per cent.}$$

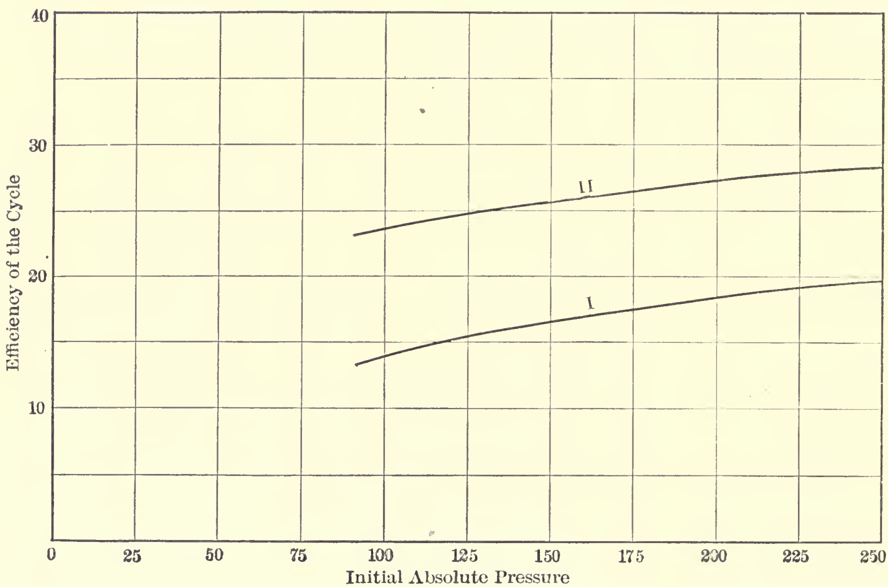


Fig. 61.—Relation between the efficiency of the Rankine cycle and the initial steam pressure. Curve I is for 15 lbs. back pressure. Curve II, is for 2 lbs. back pressure.

In order to illustrate the effect of changing the temperature or pressure range upon the efficiency of the Rankine cycle, the curves shown in Figs. 61 and 62 are drawn. The curves in Fig. 61 show the effect of varying the initial pressure for various constant back pressures, while those in Fig. 62 show the effect of varying the back pressure for various constant initial pressures. It may be noted in connection with the efficiency of this cycle, that the cycle is most efficient when dry steam is used and that when wet steam is used the efficiency gradually falls off,

as is shown by the curve in Fig. 63. The effect of increasing the superheat is shown in the same figure. That part of the diagram lying to the left of the heavy vertical line is the region of wet steam, while that lying to the right is the region of superheated steam.

155. The Rankine Jacketed Cycle. In order to minimize the loss resulting from cylinder condensation, it is often found advisable to heat the walls of the steam engine cylinder by surrounding them with a jacket or steam space, containing steam at boiler pressure. Wet steam readily absorbs heat both by conduction and radiation, while dry steam does not absorb heat readily. As the steam in the engine cylinder expands, it tends to condense and its temperature falls. The wet steam in the cylinder consequently tends to absorb heat from the cylinder walls, which in turn absorb heat from the steam jacket, so that the steam in the engine cylinder is maintained in practically a dry condition throughout the whole range of expansion, but is not, at any time, appreciably superheated. In consequence of these facts, when a steam cylinder is thoroughly jacketed, the steam within the cylinder performs a cycle, usually termed the jacketed cycle, throughout which it is assumed to remain in a dry and saturated condition. The theoretical indicator card given by an engine operating on the jacketed cycle, with complete expansion, is shown in Fig. 64. Dry steam is admitted from a to b. During expansion the steam remains in a dry and saturated condition

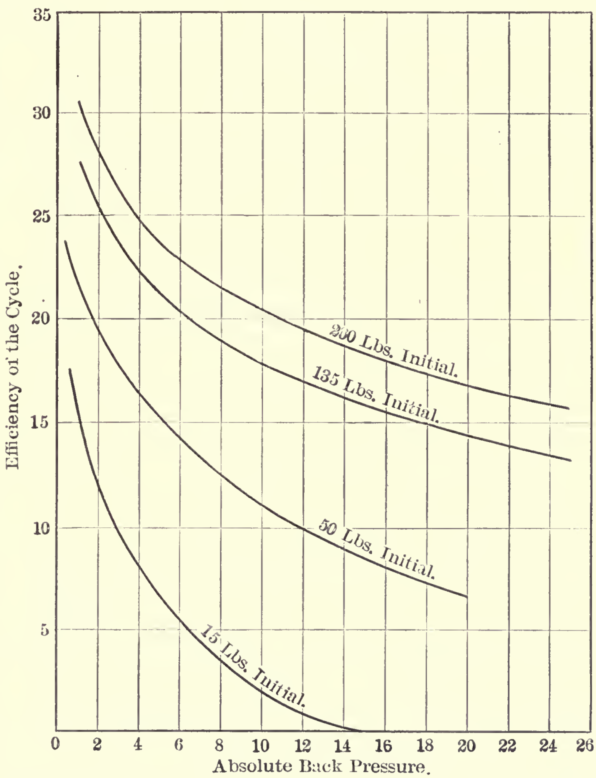


FIG. 62.—Relation between the efficiency of the Rankine cycle and the back pressure.

and line c–d is therefore a line of constant steam **weight.** The heat necessary to maintain the steam in this condition is supplied from the jacket by the liquefaction of the steam contained therein. At the end of expansion dry and saturated steam is rejected to the exhaust.

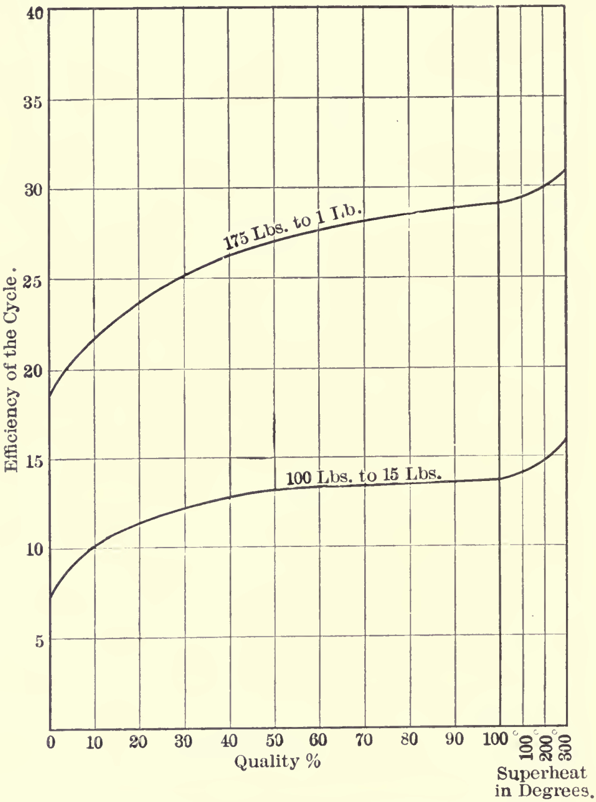


FIG. 63.—Relation between the efficiency of the Rankine cycle and the quality of the steam.

The efficiency of the Rankine jacketed cycle with complete expansion is less than that of the unjacketed cycle, as may be shown in the following manner. Referring to Fig. 65, a–b–d–c is the pressure volume diagram of the Rankine unjacketed cycle for 1 pound of steam, and a–e–f–d is the diagram of the Rankine unjacketed cycle, when the quantity of steam taken is such that the amount of dry steam in the cylinder at the end of expansion is 1 pound. b–e–f–c is then the

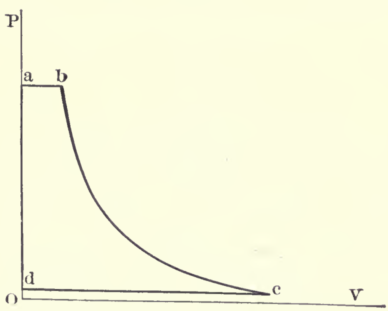


FIG. 64.—Card for a Rankine jacketed cycle.

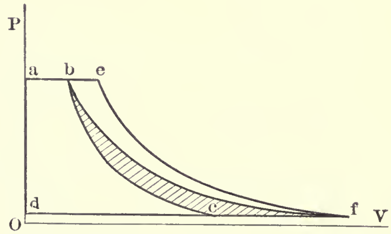


FIG. 65.—Showing the efficiency of the jacketed cycle.

equivalent of a Rankine cycle whose efficiency will be the same as the efficiency of either of the other two cycles. The line b–f is the line of constant steam weight, or the expansion line of the Rankine jacketed cycle for 1 pound of steam. The quantity of heat supplied by the jacket is equal to that represented by the area b–f–c plus the heat rejected by the Rankine cycle b–e–f–c. If we represent the heat rejected by R, the heat equivalent of the area b–f–c by J and of the area b–e–f–c by U, we will have for the efficiency of the heat supplied by the jacket,

$$E_j = \frac{J}{J + R},$$

while the efficiency of the heat supplied in the cylinder feed of an unjacketed cycle will be represented by the formula

$$E_r = \frac{U}{U + R}.$$

Since U is much larger than J, it will be seen that the efficiency of the heat supplied by the jacket is much less than if this heat had been supplied in the cylinder feed. Consequently, the efficiency of the jacketed cycle will be less than the efficiency of the unjacketed cycle.

The quantity of work done during the jacketed cycle may be determined by writing an empirical equation which expresses the relation between the pressure and volume of dry and saturated steam for the range of expansion of the cycle, and so obtaining the work done under the expansion line. The total work done during the cycle will be the work done under the expansion line plus the external work of evaporation of steam at the initial pressure, less the external work of evaporation of steam at the exhaust pressure. The heat rejected per pound of working fluid will be the latent heat of evaporation of the steam at the exhaust pressure. The heat supplied will be the sum of the heat rejected and the work done. The heat supplied from the cylinder feed will be equal to the total heat of steam at the initial pressure less the heat of the liquid at exhaust pressure. The heat supplied by the jacket will be equal to the total heat supplied less the heat supplied in the cylinder feed. The number of pounds of jacket feed per pound of cylinder feed will be found by dividing the latent heat of evaporation at the initial pressure by the heat supplied by the jacket per pound of cylinder feed.

156. Efficiency of the Jacketed Cycle with Wet Steam. In case the steam supplied to a jacketed engine is wet, the efficiency of the cycle will be seriously reduced, since the wet steam will be evaporated during the expansion period and will not perform the work which it would otherwise do. A jacketed engine in theory always rejects dry and saturated steam. Practically the steam contains a very small percentage of moisture. No theory can be developed for the jacketed cycle on the assumption

that the steam is initially wet, unless the form of expansion line is also assumed. In the case of an actual engine, it may be assumed that the expansion line has the form $PV^n = K$, and the value of the index n may be determined from the indicator card. The theory of the cycle may then be developed after finding the initial and final quality of the steam from the known cylinder volume and cylinder feed.

157. The Imperfect Cycle without Clearance. In the actual steam engine it is not practicable to expand the steam completely (i.e., to expand it till its pressure equals the back pressure) for several reasons. In the first place, it is necessary, in order to govern the speed of the engine, to have a variable terminal pressure when the load, or quantity of power developed by the engine, varies. In the next place, it will be found that the friction of the engine will be very greatly increased if the cylinder is made large enough to allow of complete expansion. In addition, there are certain losses which are increased by increasing the ratio of expansion of the steam. In order to minimize these losses, the cycle adopted in practical work is of the form already described in Fig. 29.

The effect of introducing **terminal drop** (i.e., a difference between the terminal and exhaust pressure), is of course to reduce the quantity of work performed by a given weight of steam. This may be shown graphically by the theoretical card in Fig. 66, which is the card of an engine expanding steam adiabatically from an initial pressure of 100 pounds per square inch to a final pressure of 16 pounds per square inch. The lines a–a, b–b, c–c, and d–d represent the drop in pressure at the end of the stroke, when the ratio of expansion is one, two, three, and four. The area included within the lines represents the quantity of work performed by the steam. The quantity of steam required is the same in each case, but it will be seen that the greater the ratio of expansion, the greater the quantity of work which this steam will perform. The effect of introducing terminal drop is therefore to increase the thermodynamic loss of the cycle, and to reduce the other losses in the engine. That terminal drop is chosen which makes the sum of the practical losses and the theoretical loss a minimum.

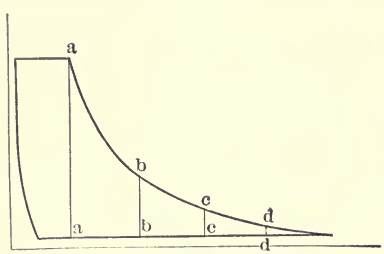


Fig. 66.—Card showing the loss due to incomplete expansion.

It may be noted that when the steam expands to a pressure lower than the exhaust pressure, not only are the actual losses still further increased, but the efficiency of the cycle is reduced. Fig. 67 is the indicator card for such a cycle. When the exhaust valve opens at point d, air or steam will rush into the cylinder from the exhaust pipe, increasing the pressure to e. This air or steam must be expelled against the back pressure.

It does no work while it is entering the cylinder, but in expelling it from the cylinder, work is done upon it represented by the area *c–d–e*. It will therefore be seen that power was lost as a result of the expansion below the back pressure.

The work done during a cycle in which there is terminal drop, but no clearance, may be found by treating the cycle as though it consisted of two parts, a Rankine cycle (area *a–b–c–f* in Fig. 68) whose back pressure is equal to the terminal pressure, and a second cycle (area *c–d–e–f*) in which the work done is equal to the product of the difference between the terminal pressure and exhaust pressure in pounds per square foot into the terminal volume in cubic feet. The terminal volume per pound of cylinder feed may be discovered by multiplying the specific volume of steam at the terminal pressure by the quality of the steam at

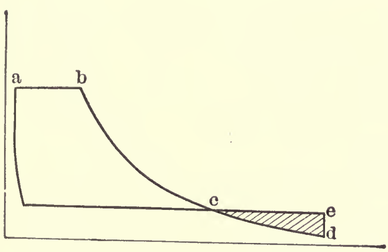


FIG. 67.—Loss due to extreme expansion.

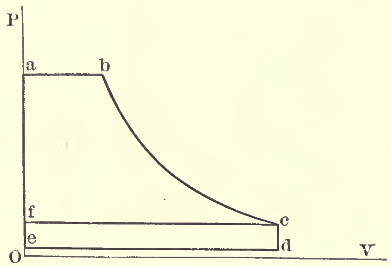


FIG. 68.—Work done during a Rankine cycle.

the end of expansion. The quality may of course be obtained from the known initial and final conditions. Designating the total heat of the steam at admission by H_1, at the terminal pressure by H_t, the terminal volume in cubic feet by V_t, the terminal pressure in pounds per square foot by P_t, the exhaust pressure by P_2, and the heat of the liquid at the temperature of exhaust by h_2, we will have for the work in foot pounds during the imperfect cycle without clearance

$$U = J(H_1 - H_t) + (P_t - P_2)V_t.$$

For the efficiency of the cycle, we will have

$$E = \frac{U}{J(H_1 - h_2)}.$$

In Fig. 69 will be found curves showing the relation between the terminal pressure and the efficiency for different conditions of initial and exhaust pressure.

158. The Effect of Clearance. It has already been shown that in case expansion and compression are complete, the efficiency of the cycle

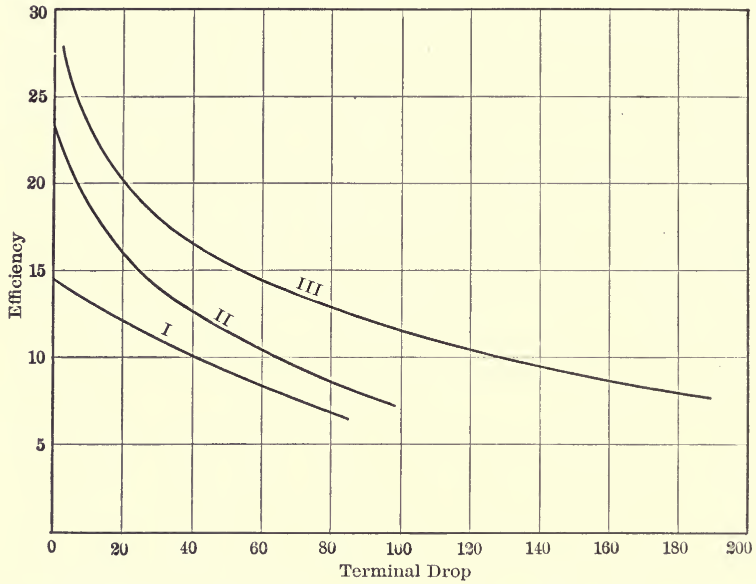


FIG. 69.—Relation between the efficiency of the imperfect cycle and the terminal drop in lbs. per sq.in.
Curve I. For 100 lbs. initial and 15 lbs. back pressure.
Curve II. For 100 lbs. initial and 2 lbs. back pressure.
Curve III. For 190 lbs. initial and 1 lb. back pressure.

is not altered by clearance. Usually, however, the efficiency of the cycle is reduced by clearance, as may be seen from the following considerations: The two simplest cases of loss from clearance are first, when the compression is complete and the expansion is incomplete, and secondly, when the expansion is complete and there is no compression. In Fig. 70 is the card of an engine having complete compression, but incomplete expansion. Every portion of the steam contained in the cylinder during expansion performs work in proportion to its mass. Consequently, the net work performed by the steam in the clearance space during expansion is represented by the area a–f–g–h. During its compression, the net work

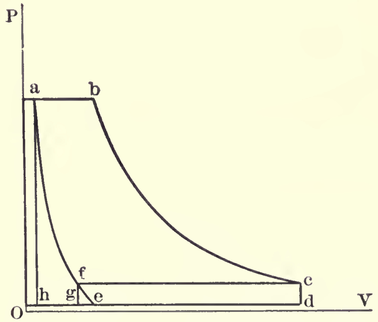


FIG 70.—Showing clearance loss with incomplete expansion.

expended upon it is represented by the area *a–e–h*. Consequently, the net loss due to the clearance is represented by the area *f–e–g*. The loss occurring in the second case may be understood by referring to Fig. 71, which is an indicator card for 1 pound of steam expanding to back pressure. Every portion of this steam performs work during the cycle in proportion to its mass. Assume that the engine has a clearance represented by the abscissa to point *b* on the diagram. The steam contained in the clearance space at the end of compression, if compressed adiabatically, would return by the path *f–a* to its initial condition. The quantity of steam introduced during admission period is represented by the volume *a–c*. All of this steam does work during expansion, but that portion of it represented by the volume *a–b* does no work during admission, except to adiabatically compress the steam already in the clearance space at point *f*. Consequently, the area *a–b–f* represents the work lost on account of clearance.

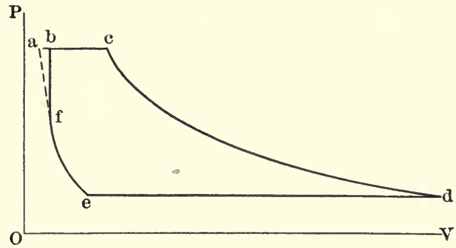


Fig. 71.—Card showing the effect of incomplete compression.

159. The Practical Cycle. In the practical cycle expansion is incomplete and we have both clearance and incomplete compression. It is therefore in order to determine the effect of these various elements on the efficiency of the cycle. Referring to Fig. 72, it may be seen that if the steam contained in the clearance space at point *f*, which is the point of compression, were compressed adiabatically to the initial pressure, it would have the volume represented by the abscissa to point *a*. The quality of the steam at *f* is, in theory, the quality which the steam would have after expanding adiabatically from its initial to its terminal pressure, since the steam which remains in the cylinder expels by its adiabatic expansion the steam which escapes from the cylinder at the instant of release. Consequently, steam compressed adiabatically from point *f* to the initial pressure would have the initial quality, which, however, would not be the quality of the cylinder feed. Were this steam compressed adiabatically, it would do the same work during complete expansion as was expended upon it during compression.

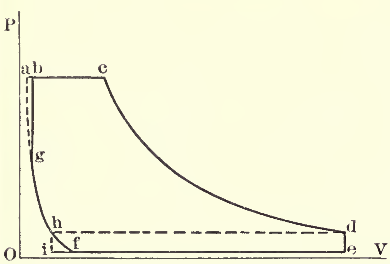


Fig. 72.—Clearance loss with incomplete expansion and compression.

However, the cushion steam expands only to the point h, whose pressure is the terminal pressure, and therefore the quantity of work expended in compressing the cushion steam exceeds the work it performs per cycle by the area f–h–i. The cylinder feed, which has the volume a–c, if working in a cylinder without clearance, would perform work represented by the area a–c–d–e–f. However, it actually performs work represented by the area b–c–d–e–f–g, and area a–b–g represents the loss on account of clearance. The sum of the areas a–b–g and h–f–i represent the total loss occurring in this cycle on account of clearance. It will be seen that the area h–f–i increases as the terminal pressure rises, and is proportional to the quantity of cushion steam. By increasing the quantity of cushion steam, we will reduce the loss represented by the area a–b–g, but we will also increase the loss represented by the area h–f–i. In theory, the best results with a given clearance are obtained when the point of compression is made such that the sum of these two losses is a minimum for the given ratio of expansion. This usually occurs when the compression is nearly complete. In practice, it is found that high compression decreases the actual efficiency of the engine, on account of its effect upon other losses.

160. Efficiency of the Practical Cycle. The determination of the theoretical efficiency of the practical cycle, on the assumption that the expansion and compression of the steam are adiabatic, is a matter of some complication. The method of computing this efficiency may be understood by referring to the card for such a cycle for 1 pound of steam, illustrated in Fig. 73. In this computation the following notation will be used: F is the weight of the cylinder feed in pounds, C is the weight of cushion steam in pounds, H_b is the total heat of the steam at point b, H_c the total heat of the steam at point c, V_c the terminal volume in cubic feet, P_c the terminal pressure in pounds per square foot, P_d the exhaust pressure in pounds per square foot, P is the pressure at point f in pounds per square foot, P_a is the initial pressure in pounds per square foot, H_e is the total heat of the exhaust steam per pound, H_f the total heat of the cushion steam per pound at point f, and h_2 the heat of the liquid at the temperature of exhaust. The weight of the working fluid, which is $F+C$, is 1 pound. The quality of the steam at point e is the quality which the steam would have in expanding adiabatically from the initial pressure and quality to the back pressure. The weight of cushion steam may be found from the known pressure, volume, and quality of the steam

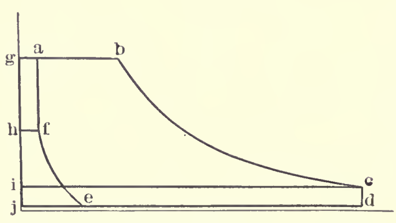


FIG. 73.—Work done by an imperfect cycle.

at point e. The total heat and other properties of the steam at point c, e and f may be computed, since the entropy of the steam is the same at these points as at b. The card may now be divided into four areas. Area g–b–c–i is a Rankine cycle for 1 pound of steam and the work done during this cycle in foot-pounds is $J(H_b - H_c)$. Area i–c–d–j is a rectangle and the work done is $V_c(P_c - P_d)$. Area h–f–e–j is a Rankine cycle performed by the cushion steam, and the work done in foot-pounds is $J\,C\,(H_f - H_c)$. Area g–a–f–h is a rectangle and the work done is equal to $V_a(P_b - P_f)$. The quantity of heat supplied during the cycle is equal to $H_b - C\,H_f - F\,h_2$. The efficiency of the cycle is found by dividing the heat equivalent of the work done per cycle by the heat supplied. In case it is desired to work out the cycle for any other quantity of working fluid than one pound, the procedure will be exactly the same except that the total heats at points b and c will be the total heats of the actual weight of working fluid.

Since all of the heat is supplied in the cylinder feed, the quality of the cylinder feed may be determined from its weight and the quantity of heat supplied. In case the quality of the cylinder feed, and its weight, and the weight of the cushion steam, are known, the total heat at c and consequently the volume and other properties of the working fluid at different points of the cycle, may be determined from the fact that $H_b = C\,H_f + F\,H_a$ in which $F\,H_a$ is the total heat of the cylinder feed. In case the volumes at b, c, and e are known and the quality of the cylinder feed is known, the weight of the cushion steam and the properties of the working fluid at different points of the cycle may be computed by successive approximation.

PROBLEMS

1. Steam operated on a Carnot cycle between pressures of 150 lbs. per square inch and 2 lbs. per square inch absolute will give what efficiency? Ans. 28.4%.

2. What will be the efficiency of steam initially dry and saturated when worked through the same pressure range in a Rankine cycle engine? Ans. 25.7%.

3. By what per cent must the total cylinder volume in Problem 2 be increased for the engine, to operate on a modified Rankine cycle and give the same power, if the clearance is 30 per cent of the total volume at cut-off? Ans. 43%.

4. Find the efficiency of a Rankine cycle engine taking steam at 90 per cent quality at a pressure of 150 lbs. absolute and rejecting it at a pressure of 2 lbs. absolute. Ans. 24.9%.

5. Find the efficiency of a Rankine cycle engine taking steam at 150 lbs. absolute and 200° superheat and rejecting it at a pressure of 2 lbs. absolute. Ans. 33.4%.

6. A jacketed engine without clearance and with complete expansion takes dry and saturated steam at 100 lbs. absolute, and rejects dry and saturated steam at 16 lbs. absolute. Find the index of the expansion line. Ans. 1.152.

7. Find the constant of the above curve for 1 lb. of steam.

Ans. $K = 487$ when p is in pounds per square inch and V in cubic feet.

8. Find the work done during expansion by 1 lb. of steam.
Ans. 104,200 ft.lbs.

9. Find the heat added during expansion. Ans. 108.1 B.T.U.

10. Find the work of the cycle. Ans. 110,800 ft.lbs.

11. Find the efficiency of the cycle. Ans. 12.80%.

12. Find the efficiency of a Rankine cycle engine working dry and saturated steam through the same pressure range. Ans. 13.4%.

13. An engine without clearance takes 1 lb. of steam of 90 per cent quality and 100 lbs. pressure, and expands it adiabatically to 25 lbs. pressure. Find the terminal volume. Ans. 14.96 cu.ft.

14. The back pressure in the above problem is 2 lbs. Find the work done during the cycle. Ans. 148,060 ft.lbs.

15. Find the efficiency of the cycle. Ans. 19.0%.

16. If the clearance space in the above engine is made equal to 50 per cent of the swept volume to the point of cut-off, and compression is complete, what will be the swept volume for 1 lb. of cylinder feed? Ans. 20.23 cu.ft.

17. What will be the work done per pound of steam supplied?
Ans. 120,290 ft.lbs.

18. What will be the efficiency of this cycle? Ans. 14.4%.

19. An engine takes steam of 100 lbs. pressure. The steam is dry and saturated at cut-off, and has a volume of 4.429 cu.ft. It expands adiabatically to 30 lbs. pressure. The back pressure is 15 lbs. per square inch. The volume of the cushion steam at the point of compression is 4 cu.ft. The pressure at the end of compression is 60 lbs. What is the clearance volume? Ans. 1.185 cu.ft.

20. What is the volume at release? Ans. 12.74 cu.ft.

21. What is the weight of the cushion steam? Ans. .171 lb.

22. What is the work done during the cycle? Ans. 78,720 ft.lbs.

23. What is the weight of the cylinder feed? Ans. .829 lb.

24. What is the total heat of the cylinder feed? Ans. 990.3 B.T.U.

25. What is the heat supplied in the boiler to each pound of cylinder feed?
Ans. 1016 B.T.U.

26. What is the efficiency of the cycle? Ans. 12.0%.

CHAPTER X

LOSSES IN THE STEAM ENGINE

161. Classification. The losses which occur in the steam engine ma be classified under nine heads, as follows:

1. Unavoidable thermodynamic loss.
2. Losses due to the imperfection of the cycle employed.
3. Losses due to the imperfection of the condensing machinery employed.
4. Losses due to wire drawing and fluid friction.
5. Losses due to cylinder condensation.
6. Losses due to valve and piston leakage.
7. Losses due to the conduction and radiation of heat.
8. Losses due to clearance.
9. Losses due to mechanical friction.

162. Loss when a Perfect Cycle is Employed. The first of these losses cannot be reduced by any method whatever, without changing the temperature range through which it is possible for the working fluid to operate. It is the thermodynamic loss of the Carnot cycle. Expressed as a fraction of the total heat supplied this loss is equal to $\frac{T_2}{T_1}$, in which T_1 is equal to the absolute temperature of the steam supplied to the engine, and T_2 is the temperature of the circulating water discharged from the condenser. In the case of an engine using saturated steam, the unavoidable thermodynamic loss is fixed on the one hand by the highest steam pressure which it is safe and profitable to carry, and on the other hand by the quantity and temperature of the condensing water available. In case superheated steam is employed, the upper limit of temperature may be considerably raised.

163. The Practical Limits of Pressure and Superheat. The upper limit of steam pressure is usually found to be between 200 and 250 pounds per square inch, the corresponding temperature being from 380 to 400° F. Engines operating at higher pressures than this usually give trouble; difficulties are encountered in properly constructing and maintaining the boilers and pipe lines, and the increased dangers and expense of operation more than overbalance the resulting gain in thermodynamic efficiency. We may therefore place the upper limit of the temperature range at 400° F. or 860° absolute in the case of engines using saturated steam.

When superheated steam is employed, the upper limit of the temperature range may be raised to about 600° F. At higher temperature than this, lubrication of the cylinders and valves of a reciprocating engine becomes impossible, and on account of excessive expansion and weakening of materials of construction, difficulties begin to be encountered in steam turbine operation. The upper limit of temperature with superheated steam is therefore about 1060° absolute.

164. Lowest Practicable Temperature of Condensation. In practice the final temperature of the condensing water depends on the quantity of condensing water available, and on its initial temperature. It is, however, practically impossible to secure a final temperature lower than 70° F., except in winter, or when a large supply of cool condensing water is available. In summer, and especially in the tropics, the final temperature of the condensing water will rise to 100 or 110° F. The lower limit of temperature range is therefore about 530 to 570° absolute.

165. The Per Cent of Unavoidable Loss. Since the extreme temperature range practicable for steam engines is from 1060° absolute to 530° absolute, the unavoidable thermodynamic loss can never be less than 50 per cent. Except under the most favorable conditions it is, extremely difficult to realize a temperature range greater than from 960° to 560° absolute, under which conditions 41.6 per cent of the heat supplied is available for transformation into work and 58.4 per cent is unavoidably lost. In case saturated steam is used, the extreme range will rarely be greater than from 860 to 560° absolute, in which case 35 per cent of the heat is available, and 65 per cent is unavoidably lost. When the engine is run non-condensing with a steam pressure of 100 pounds absolute, only about 14.7 per cent of the heat supplied is available, and the unavoidable thermodynamic loss is 85.3 per cent. Compound condensing engines are usually operated under such conditions that only about 15 to 25 per cent of the heat supplied is available. It will thus be seen that of the heat supplied to a steam engine, from 50 to 85 per cent is unavoidably lost, even when the engine is ideally perfect in every detail.

166. Loss Due to Imperfection of Cycle. The efficiencies of the cycles commonly employed in steam engine work have already been discussed at length in Chapter IX. The loss due to the imperfection of the cycle employed, expressed as a per cent of the total heat supplied, is equal to the difference between the efficiency of the perfect cycle and that of the imperfect cycle actually employed. It is better, however, to express this loss as a per cent of the available heat. We may do so by dividing the difference between the efficiency of the perfect cycle and the imperfect cycle, by the efficiency of the perfect cycle. This loss is minimized by adopting the most efficient cycle possible.

167. Effect of Imperfect Condenser Action. The effect of the third source of loss is to raise the temperature and pressure of the steam entering the condenser. If it were possible to bring the condensing water and the steam together in such a way that they would attain a common temperature, and at the same time not introduce air into the condenser, the action of the condenser would be perfect. However, in order to condense the steam, it is necessary that the final temperature of the circulating water be somewhat less than that of the condensing steam. The required temperature difference is variable, amounting sometimes to over 20°. In addition, air is present in the condenser, and its presence prevents the pressure of the steam in the exhaust pipe from reaching the pressure, and therefore the temperature, of the steam in the condenser itself. On account of the presence of air and the imperfect cooling of the steam, the temperature range of the working fluid, and therefore the efficiency of the engine, is reduced. The loss from this source depends upon the efficiency of the cycle employed, becoming greater as the efficiency of the cycle increases, hence the importance of good condensing machinery in connection with steam engines and turbines of high efficiency. The amount of this loss may be determined by computing the theoretical efficiency of the cycle employed, at the observed back pressure (call this efficiency E_0), and at the back pressure corresponding to the temperature of the discharged circulating water (call this efficiency E_t), and taking their difference (which is $E_t - E_0$). The result will be the amount of this loss expressed as a per cent of the heat supplied. It would be more proper, however, to express it as a per cent of the total heat transformable into work by the cycle employed, which may be done by dividing the difference found above by the quantity E_t.

168. Loss from Wire Drawing and Steam Friction. The fourth source of loss in the steam engine arises from the fact that a difference of pressure is necessary in order to force the steam through the port openings and steam passages of the engine at the necessary velocity. The loss in pressure incurred in forcing steam through a restricted port opening at high velocity is said to be due to **wire drawing.** The loss in pressure incurred on account of the roughness and crookedness of the steam passages is said to be due to **fluid friction.** In each case the loss in pressure is approximately proportional to the square of the velocity of the steam. When these openings and passages are of ample area, so that the maximum velocity of the steam does not exceed 6000 to 8000 feet per minute, the loss of pressure is very small. When, however, the passages are restricted, and the valves do not open and close promptly, the pressure difference becomes considerable and the area of the card actually given by the engine is materially less than that of the theoretical card which would be given by the engine in case its ports were ample,

and its valves opened and closed instantly. The ratio of the area of the actual card to the area of the theoretical card is termed the **card factor of the engine,** and is usually expressed as a per cent. The loss due to fluid friction and wire drawing, expressed as a per cent of the work represented by the area of the theoretical card, is found by subtracting the card factor from 100 per cent.

169. Values of the Card Factor. The card factor for locomotives is usually from 75 to 85 per cent. For ordinary high speed engines with ample ports the card factor will range from 85 to 95 per cent. The card factor for a good Corliss engine is about 95 to 98 per cent, while for slow-moving pumping engines equipped with Corliss valves the card factor is practically 100 per cent. It will be seen that this loss varies from about 25 per cent to less than 1 per cent. It may be reduced by making the steam and exhaust ports short and direct, and of ample area, and so operating the valves that they open and close promptly.

170. The Design of Engine Ports. The ports of steam engines are usually designed by making the cross-sectional area of the inlet passages such that the nominal velocity of the steam through them is from 5000 to 9000 feet per minute, and the area of the exhaust passages such that the nominal velocity of the steam through them is from 4000 to 7000 feet per minute. The nominal velocity of the steam is found by dividing the piston area by the port area and multiplying the quotient by the mean piston speed. Consequently, the formula for the design of ports will be

$$P = \frac{S\,A}{V},$$

in which P is the port area;
A is the piston area;
S is the mean piston speed;
V is the nominal steam velocity.

Since the actual piston speed is variable, and since some of the steam supplied condenses in the cylinder during admission, the actual velocity of the steam in the inlet and exhaust passages is from 50 to 75 per cent higher than the nominal velocity, during some parts of the stroke. Most types of valves open and close gradually and therefore greatly restrict the port openings during a considerable portion of the stroke. The effect of this restriction is to very greatly increase the loss due to wire drawing. It is impossible to estimate the loss from this source, except by making comparisons with engines in which this loss has been measured.

171. Cylinder Condensation. The most important source of loss in the steam engine is due to the condensation of steam upon the cylinder

wall during the admission period, and its subsequent evaporation during the period of expansion and exhaust. The cause of this loss will become apparent when we consider the phenomena which occurred in the cylinder while the engine is in operation. The wall which encloses the working fluid is of cast iron or steel, and is therefore a good conductor of heat. It has been shown both from theory and by actual measurement that the surface of this wall, at the instant of admission, is somewhat cooler than the entering steam. On account of this difference in temperature, at the instant of admission some of the steam immediately condenses upon the wall surface,[1] raising its temperature. Since the wall is a good conductor, heat begins to flow from the surface into the wall. Were it not for this flow of heat the temperature of the surface would be instantly raised to that of the entering steam, and condensation would cease as soon as it began. On account of this flow of heat, the temperature of the surface cannot be raised instantly to that of the steam in contact with it, and the condensation goes on at a gradually decreasing rate throughout the period of admission.

When expansion begins, the temperature of the steam begins to fall, and finally it becomes equal to the temperature of the wall surface. At this instant condensation ceases. After this point is passed, the temperature of the wall surface is greater than that of the expanding steam, and the moisture which has condensed upon it begins to evaporate, the heat now flowing to the surface from the interior of the wall. As the steam pressure continues to fall, the evaporation becomes so rapid as to be almost explosive in character. In consequence of this the steam evaporated from the wall during the latter part of the expansion period is quite wet. If the steam is not all evaporated by the end of the expansion period, and probably it usually is not, the remainder of the water is blown off in the form of spray at the instant that the exhaust valve opens and terminal drop occurs.

During the exhaust stroke the wall surface is much hotter than the steam contained in the cylinder, and therefore the steam is dried by the heat radiated to it from the wall. At the beginning of compression the layer of steam in immediate contact with the wall is superheated. However, since superheated steam absorbs heat with difficulty, the radiant heat from the wall is unable to superheat the main body of cushion steam to any appreciable extent, and this steam is, at the beginning of compression, practically dry and saturated.

[1] It is on this account that it is necessary to open the inlet valve before the engine begins its working stroke. If the valve is not so opened, the pressure in the cylinder will not begin to rise until the piston has completed a portion of the working stroke, and there will be a considerable loss in power without any change in the steam consumption of the engine.

After the closing of the exhaust valve, as a result of adiabatic compression the whole of the cushion steam is superheated, and its temperature raised above the temperature of the cylinder walls. As soon as its pressure becomes that corresponding to the temperature of the walls, this superheated steam begins to condense, exactly as moisture from the air condenses upon the surface of a cold object whose temperature is below the dew-point. In a great many engines, compression is not carried to this point, but in high speed engines with light load, the compression is often sufficient to show this phenomenon by a sudden change in the direction of the compression line on the indicator card. This effect may be noted at *a* in Fig. 74, where there is a decrease in the volume of the cushion steam without a corresponding change in pressure.

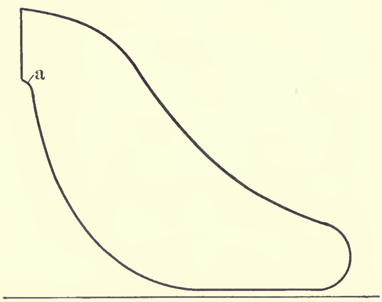


FIG. 74.—Card showing cylinder condensation during compression.

172. The Amount of Heat Interchanged. Since the steam alternately imparts heat to, and extracts it from, the cylinder wall, the temperature of the wall surface, and consequently of every point within the wall, undergoes periodic variation. This temperature variation is a maximum at the wall surface, and grows rapidly less as the distance from the particle to the wall surface is increased. The amount of the temperature variation is shown by Professor Cotterill[1] to be given by the equation

$$R = R_s e^{-mx}, \qquad \ldots \ldots \ldots \quad (1)$$

in which R is the temperature range of any particle, R_s is the temperature range at the wall surface, x is the distance of the particle from the surface, and m is determined by the thermal properties of the material of the wall and the periodicity of the cycle. Its value is given by the equation

$$m = \sqrt{\frac{\pi N w s}{f}}. \qquad \ldots \ldots \ldots \quad (2)$$

In which N is the number of cycles per second, w is the density of the metal in pounds per cubic foot, s is the specific heat of the metal, and f is the specific conductivity of the metal in B.T.U. per second per cubic foot, per degree difference in temperature.

As a result of the periodic variation in temperature of the wall surface, a definite quantity of heat is imparted to each square foot of the

[1] See Chapter X of Cotterill's work, "The Steam Engine."

wall by the steam, and again rejected by the wall to the steam, during each revolution of the engine. This quantity of heat may be shown to be

$$Q = K R_s \sqrt{\frac{2 w s f}{N}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (3)$$

in which Q is the number of B.T.U. surrendered by the steam to each square foot of the wall surface per cycle, K is a constant depending upon the form of the temperature cycle of the wall surface, R_s is the temperature range of the wall surface, and w s f and N are as in the preceding paragraph. The value of the constant K is unity in case the temperature cycle of the wall surface is harmonic, and is very nearly unity for other probable forms of the cycle.

173. The Practical Aspects of Cylinder Condensation. It might be thought that the loss due to cylinder condensation could be deduced directly from equation (3) in the preceding article. This would be true if the temperature range of the wall surface were known. However, the temperature range depends on the form of the indicator card, the pressure range of the steam, the quality of the steam entering the cylinder and the rotational speed of the engine, and it is obviously impossible to determine it with any accuracy. Hence we can make only a rough estimate of the probable amount of cylinder condensation in the case of any particular engine, operating under given conditions. We may, however, readily determine what changes are necessary in the operating conditions in order to minimize the cylinder condensation. An inspection of the equation will make it apparent that the loss may be reduced: first, by reducing the temperature range of the wall surface; second, by increasing the rotational speed of the engine; third, by reducing the area of the wall surface enclosing the clearance space; and fourth, by making the wall of non-conducting material.

174. Methods of Reducing the Temperature Range of the Wall Surface. Five methods are available for reducing the temperature range of the wall surface: first, by increasing the rotational speed of the engine; second, by supplying the engine with dry or superheated steam; third, by decreasing the ratio of expansion; fourth, by reducing the temperature range of the steam in the cylinder; and fifth, by jacketing the cylinder with steam of boiler pressure. The effect of increasing the rotational speed of the engine is to increase the rate at which the pressure falls during expansion, to increase the rapidity of evaporation, and consequently the wetness of the steam evaporated during the expansion period, to reduce the heat loss from the wall due to this re-evaporation, and therefore to reduce the temperature range of the wall surface.

The effect of supplying an engine with wet steam is to increase the quantity of water deposited upon the wall surface by a given heat transfer,

to increase the loss of heat due to the subsequent re-evaporation of this water, and therefore to increase the temperature range of the wall surface. By supplying the engine with dry or superheated steam, the heat loss from the wall caused by the re-evaporation of the deposited moisture is greatly diminished, and the temperature range of the wall surface correspondingly reduced. The greater the superheat of the steam supplied the less the loss from re-evaporation, and therefore the less the temperature range of the wall surface.

Decreasing the ratio of expansion reduces the heat loss during the expansion period by shortening this portion of the cycle. It also reduces the heat loss by increasing the terminal drop and so removing the moisture more completely at the instant of release, by its explosive evaporation.

Other things being equal, it is apparent that the temperature range of the wall surface must be proportional to the temperature range of the steam in the cylinder. We may reduce the temperature range of the steam in the cylinder and also the ratio of expansion, by using a multiple expansion engine. In the case of a compound engine, the temperature range of the steam is reduced to one-half, and in the case of a triple expansion engine to one-third of its value for a simple engine of the same pressure range. The cylinder condensation is reduced by a still larger amount, since the ratio of expansion is also reduced.

The effect of a steam jacket is to raise the mean temperature of the cylinder wall and therefore to reduce the initial condensation. Since less steam is condensed, less heat will be lost by its re-evaporation, and the effect of the jacket is therefore to greatly reduce the temperature range of the wall surface. A steam jacket properly applied always increases the efficiency of an engine, since the thermodynamic loss due to the use of the jacket is always less than the reduction effected by the jacket in the loss from cylinder condensation. However, the effect of the jacket in increasing the economy of large engines of high piston speed is insignificant, and the cost of applying the jacket to such engines does not warrant the slight saving resulting from its use. Jackets are therefore not usually applied to engines having a rotational speed of more than 60 to 75 revolutions per minute, unless they are desirable for operating reasons, as, for instance, to enable the engineer to quickly warm up the engine when starting.

175. Effect of Increasing the Speed of Rotation. It has already been shown that increasing the rotational speed of the engine affects the amount of cylinder condensation indirectly by reducing the temperature range of the wall surface. An inspection of Equation (3) Art. 172, will show also that it affects this loss directly, the amount of the loss for a given temperature range being inversely proportional to the square root of the number of revolutions per minute. On both of these accounts it is

desirable that an engine should be operated at a high rotational speed. The general design of high speed engines, however, is usually such as to make them very wasteful of steam on account of the type of valve employed, the large clearance volume, the heavy compression, and the great area of clearance surface. In practice the greatest steam economy is obtained from an engine of long stroke and high piston speed, but of comparatively low rotational speed.

176. Reducing the Clearance Area. The area of the wall surface upon which cylinder condensation occurs may be reduced by making the steam and exhaust ports short and direct and by giving the engine a high piston speed. The high speed automatic engine usually has long and crooked ports, and being of very short stroke has a low mean piston speed. Consequently, the loss from cylinder condensation is greater in such engines than it is in long-stroke four-valve engines of equal power, which have short, direct ports and high piston speed.

It must be borne in mind that it is not the cylinder condensation **per cycle** which the designer should seek to minimize, but rather the cylinder condensation **per pound of steam supplied.** Increasing the rotational speed of an engine by shortening the stroke, and without changing the piston speed, will reduce the weight of cylinder feed per revolution in greater ratio than it reduces the weight of cylinder condensation per revolution, and will therefore increase the loss from condensation. For mechanical reasons it is advisable to limit the mean piston speed to 1000, or at the utmost 1200 feet per minute, while speeds of 600 to 900 feet are usually employed. The length of stroke is determined by financial considerations, long-stroke engines being more expensive for a given power than short-stroke engines. In practice the stroke is usually limited to three times the diameter of the high pressure cylinder, and is rarely greater than 6 feet.

177. The Use of a Non-Conducting Wall. The application of non-conducting materials to those parts of the clearance surface that are not subject to wear has often been advocated. However, actual tests of engines in which the clearance surfaces have been covered with porcelain, glass, slate or other non-conducting materials, have not usually shown sufficient gain in economy to warrant the use of this method of reducing cylinder condensation. In general, such tests have shown but little increase in economy, although Thurston has reported a reduction of 60 per cent in the amount of cylinder condensation from the use of this method. It is highly probable, however, that the steam jacket is a more efficient and practical method of reducing the loss.

178. Weight of Steam Condensed Per Revolution. It is apparent from the foregoing discussion of cylinder condensation that it is impossible to derive a rational formula which will give accurately the amount of

steam condensed per revolution when the dimensions of the engine, the conditions of operation, and the form of indicator card are known. However, a large number of empirical equations have been developed by which this quantity may be determined with more or less accuracy.* An investigation of a large number of engine tests serves to show that the amount of cylinder condensation may be determined approximately by the equation

$$C = .00033\, A\left(\frac{R_x + R}{N^{.588}}\right),$$

in which C is the number of pounds of steam condensed per revolution,

A is the number of square feet of wall surface exposed per revolution to the action of the steam at cut-off,

R_x is the temperature range of the steam during the expansion period in degrees Fahrenheit,

R is the total temperature range of the steam, and

N is the number of revolutions per minute.

The application of this formula will be readily understood from the following example: An engine having a 12″×36″ high pressure cylinder takes steam at 160 pounds absolute. The ratio of expansion is 3 and the back pressure is 30 pounds absolute. The area of wall surface exposed to the action of steam at cut-off is 16 square feet. The number of revolutions per minute is 125. Required the weight of steam condensed per revolution.

Assuming hyperbolic expansion, the pressure at release will be one-third the initial absolute pressure, or 53 pounds per square inch. The temperature of the steam at admission is 364°, at release 285° and at the back pressure 250°. The temperature range during expansion is 79° and the entire temperature range is 114°. Substituting in the formula we will have

$$C = .00033 \times 16\left(\frac{79 + 114}{125^{.588}}\right) = .069 \text{ lbs.}$$

The weight of steam condensed per stroke is therefore about .035 pound. By adding this quantity to the cylinder feed per stroke shown by the card, the probable weight of steam consumed per stroke of the engine may be computed.

In computing the area of the wall surface exposed to the action of the steam per revolution at cut-off, it is necessary to divide this surface into three parts. The first part is the area of the cylinder head and the piston. The second part is the area of the walls enclosing the ports and steam passages, together with the valve faces. The third part is the surface of the cylinder barrel up to the point of cut-off. The first and third parts may be computed from the general dimensions of the engine,

while the second part must be computed from the detail drawings of the engine cylinder. The areas must be computed for both the head and crank end of the cylinder and their sum taken, when the steam condensed per revolution is desired.

179. Valve and Piston Leakage. The importance of the sixth source of loss in the steam engine will depend upon the design, the workmanship and the method of operation of the engine. The leakage past the piston of a steam engine ought to be very slight, in practice, if the piston is properly made and provided with properly fitted packing rings. It is usual to make the piston from .005 to .015 inch smaller in diameter than the cylinder, and then to provide the piston with two elastic packing rings which expand and prevent the escape of steam. In case these rings are broken, or lose their elasticity, the loss from piston leakage may become a considerable quantity, but this rarely happens when the engine receives proper care.

A valve which is forced down upon its seat by an unbalanced steam pressure will be steam tight after it has worn to a good bearing. When such a valve is new, however, although the surfaces of the valve and seat may be scraped to an exact plane while cold, the valve will not necessarily be tight when it is hot. The heat and the pressure of the steam upon the back of the valve invariably tend to distort these surfaces. The high spots soon wear down, however, and the valve becomes tight. A Corliss valve, like a slide valve, tends to wear tight. It will be seen, therefore, that a plain slide valve, a Corliss valve, and a properly fitted poppet valve, will not leak under service conditions.

Piston valves and balanced slide valves, on the other hand, are practically certain to leak. In order that the valve shall slide freely, it is necessary that the distance between the balance plate and the valve seat shall exceed the thickness of the valve by from 0.003 inch to 0.005 inch. In the case of a piston valve, a similar difference is required between the diameter of the valve and the valve seat. In consequence of this fact, when such a balanced valve is reciprocated, there is a very considerable space through which steam and water may find their way directly from the steam chest to the exhaust port. No definite value can be set for the amount of this loss, since it will depend on the clearance of the valve, on the value of the steam and exhaust pressure, on the lap of the valve, and on the kind and amount of lubricant used on the valve. The amount of this leakage per revolution is often equal to or greater than the amount of cylinder condensation per revolution.

180. Effects of Leakage upon the Indicator Card. The effect of valve and piston leakage upon the indicator card of a steam engine is exactly the same as that of cylinder condensation. A leak past the piston, or from the cylinder into the exhaust, during the admission period,

gives exactly the same effect as does initial condensation. A leak from the steam chest into the cylinder during the expansion period gives the same effect as re-evaporation. There is no way by which the effects of cylinder condensation and re-evaporation may be separated from those of leakage in the case of an engine test, or by which the amount of either may be determined, and they must therefore be considered together in an analysis of such a test. However, since the two kinds of losses arise from entirely different causes, it is necessary to consider them separately when attempting to reduce them by correct methods of engine design.

It may be pointed out in this connection that while it is possible to measure the amount of leakage while an engine is blocked in a given position, it is impossible to measure or to estimate the leakage which occurs in that engine under operating conditions. If an engine is blocked in position and steam is turned on, it will usually be found that no important leak takes place either through the valves or past the piston. If the valves of this engine be then made to move without uncovering the ports, they will be found to leak. If the ports of a slide valve engine be blocked by some means, for instance by filling them with lead, and the valve be made to move in the normal manner, a considerable leak will usually be discovered from the steam chest into the exhaust port. A part of this is steam leakage, while a part is due to condensation and subsequent evaporation of the steam upon the valve surface, the ports, etcetera. Measuring the leakage under these conditions will not, however, determine the leakage under normal operating conditions, since the quality and amount of steam passing through the valve ports is radically different.

It will be seen that the automatic engine with a balanced slide valve is essentially wasteful, on account of this source of loss. This is one reason for its rapid displacement of late years by the four-valve automatic engine, in which this source of loss is greatly reduced, if not entirely obviated. There is no possible way by which the loss due to leakage in a Corliss or other four-valve engine can be measured, but it is probable that this loss under running conditions is not materially greater than it is when the engine is blocked and the valves are closed.

181. Conduction and Radiation. The seventh source of loss, namely the conduction and radiation of heat from the steam cylinder, has the effect of increasing the cylinder condensation by reducing the mean temperature of the cylinder wall and thereby increasing the temperature range of the wall surface. Its amount in the case of a jacketed engine may be measured by determining the quantity of steam condensed in the jacket when the engine is not running. In the case of unjacketed engines the amount of this loss cannot be determined, and the extent of its effect upon cylinder condensation is impossible of estimation. This

source of loss may be very largely eliminated by covering the exterior surface of the cylinder with some non-conducting material, such as mineral wool, asbestos sponge, or magnesia. This non-conducting coating is usually covered by a lagging of cast iron or sheet steel for the purpose of protecting it from mechanical injury and improving the appearance of the engine.

182. Losses Due to Clearance. It has already been shown in the preceding chapter that, when an ideal engine has clearance no thermodynamic loss results if the expansion and the compression are complete. In the case of a practical engine, there is always a loss due to clearance, even though both the expansion and compression are complete. The work of compressing the cushion steam in such an engine is much greater than the work done by the cushion steam during expansion, since the steam is practically dry and saturated at the beginning of compression, while it is quite wet at the beginning of expansion, on account of initial condensation. It will be seen that the amount of this loss depends on the weight of the cushion steam and can be reduced only by reducing the clearance volume and the compression pressure to a minimum. In case no compression is employed the theoretical efficiency of the cycle will be reduced. It is therefore advisable from the standpoint of efficiency to adopt that degree of compression which will make the sum of the theoretical and the practical losses a minimum. The greater the amount of cylinder condensation the greater will be the relative importance of the practical loss and the less the degree of compression which may be profitably employed. It will, therefore, be found in practice that compression is undesirable when the amount of cylinder condensation is large.

With some types of engines, it is unadvisable for mechanical reasons to do away with compression, or even to reduce it very much. This is the case with all high-speed engines. A considerable amount of compression and a large clearance space is necessary in order that such engines shall operate smoothly, and without excessive depreciation. The purpose of the cushion steam is to take up the shock which would otherwise be experienced as a result of the rapid reversal of the heavy reciprocating parts at the end of the stroke. High speed engines are necessarily made with large clearance, in order that there shall be sufficient cushion steam to serve this purpose. It will therefore be seen that the clearance losses are high in this type of engine, and it is partly on this account that the slow-moving long-stroke engine with its small clearance will usually be found to be more efficient.

183. Friction Losses. Usually from 6 to 14 per cent of the power developed in the cylinder of an engine at rated load is lost in overcoming the friction of the moving parts. The amount of this loss varies with the speed of the engine, but is practically independent of the load. Con-

sequently, the indicated power developed when the engine is running idle measures the amount of this loss. This loss may be minimized by the proper design and lubrication of the bearings and by careful attention to their adjustment and alignment. The amount of the loss depends upon the weight of the moving parts, the relative velocity of the rubbing surfaces, the quality of the lubricant, the method of introducing the lubricant, the fit of the bearings, and the ratio of the maximum to the mean effective pressure. Heavy moving parts, by increasing the pressure on the bearings, increase the friction loss. When the shafts and pins are larger in diameter than is necessary for proper strength and stiffness, the friction loss is increased on account of the higher velocity of rubbing. When a copious supply of good lubricant is furnished in such a manner that it completely lubricates the rubbing surfaces, the loss is reduced. The most efficient method of accomplishing this is to furnish an excess of the lubricant at the points where the bearing pressure is the greatest, by means of a force pump, so that the shaft is practically floated on oil. When the sup-

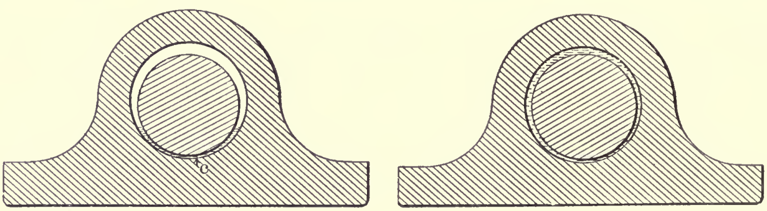


FIG. 75 *a*. FIG. 75 *b*.
Showing the effect of excessive running clearance.

ply of oil is insufficient the lubricating film between the rubbing surfaces is thin. The amount of friction loss varies inversely with the thickness of this film, hence the desirability of an ample supply of lubricant. The difference between the diameter of the shaft or pin, and of the box in which it rotates, is an important matter. If the difference is too small, the lubricating film is necessarily thin, and the friction loss high. If the difference is too great, the shaft will be supported in the manner shown in Fig. 75 *a*, instead of that shown in Fig. 75 *b*, and the lubricating film will be easily destroyed on account of the excessive pressure along the narrow surface of contact at *c*.

When the ratio of the maximum to the mean effective pressure is high, not only must the moving parts be heavy in order to resist the excessive stresses imposed at certain parts of the stroke, but the pressure transmitted from the piston to the bearings will be large in comparison with the power actually developed. A low ratio of expansion is therefore favorable to high mechanical efficiency. The mechanical efficiency of a compound engine will be higher than that of a simple engine having

the same total expansion, since the expansion is divided between two cylinders in the case of a compound engine, and the ratio of the maximum to the mean effective pressure is greatly reduced. By increasing the number of cylinders acting upon a shaft, the turning moment is made more even, and the weight of the fly-wheel may be reduced. A cross compound engine in which two cylinders act on crank pins set at right angles is therefore more efficient mechanically than a single cylinder engine or a tandem compound engine of the same power and speed.

Since the fly-wheel and other moving parts of a high speed engine are usually much lighter than those of a long stroke engine of the same power, and since the ratio of expansion is greater in the case of a long stroke engine, a high speed engine is usually mechanically more efficient than a long stroke engine. A few tests are on record which indicate an exceedingly small loss from mechanical friction in high speed engines, sometimes as low as 2 per cent of the indicated horse-power of the engine, but it is doubtful whether these unusual results were realized, or whether the tests were inaccurate.

184. Reducing the Losses by Proper Design. It will be noted that some of these losses are of such a nature that when one is decreased, another one is increased. It should be the aim of the engine designer to reduce the sum of the losses to a minimum, which involves balancing these losses one against another. For instance, increasing the ratio of expansion reduces the loss due to the imperfection of the cycle and increases that due to cylinder condensation. For some particular ratio of expansion, the sum of these two losses will be a minimum, and that ratio of expansion will be the one chosen.

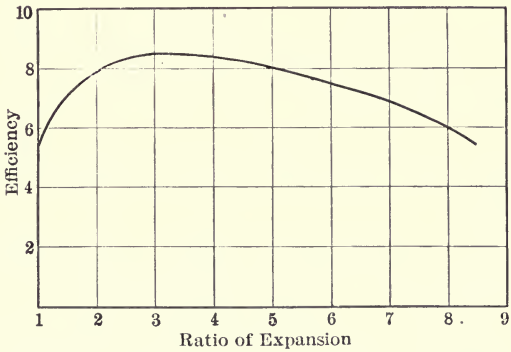


Fig. 76.—Effect of changing the ratio of expansion on the efficiency of an engine.

The design of an engine, however, is not an exact process in which the various losses are exactly estimated and balanced one against another, and certain dimensions accurately determined which will make the sum of these losses a minimum. On the contrary, a considerable latitude may be allowed in fixing upon the principal dimensions of an engine without affecting its efficiency to any noticable degree. In Fig. 76 will be found a curve giving the relation of the ratio of expansion occurring in an actual engine to the efficiency of the engine. It will be seen that

as the ratio of expansion is increased the efficiency rises and then falls off, and that for the ratio of expansion between $2\frac{3}{4}$ and 4, the efficiency is practically constant. Similar effects may be noticed in regard to changes in almost every one of the principal dimensions of an engine.

Not all of the sources of loss are of this character, however. Increasing the clearance volume of an engine, for instance, always reduces the efficiency of the engine. Increasing the clearance area invariably has the same effect. In cases of this kind, it should be the aim of the designer to adopt every expedient which will reduce such losses to a minimum.

PROBLEMS

1. What per cent of the heat supplied is unavoidably lost with the perfect cycle when steam is supplied at 100 lbs., absolute and exhausted at a pressure of one atmosphere? Ans. 15 per cent.

2. What per cent is unavoidably lost when steam is supplied at 180 lbs. absolute and a superheat of 200° and the final temperature of the condensing water is 90° F.? Ans. 53.2 per cent.

3. If the theoretical efficiency of the cycle employed in the first case is 10 per cent, what per cent of the available heat is lost? Ans. 33 per cent.

4. A Rankine cycle is employed in Problem 2. What per cent of the available heat is lost? Ans. 33.4 per cent.

5. If the back pressure in the engine in Problem 4 be 1.5 lbs. instead of that corresponding to the temperature of the discharged condensing water, what per cent of power will be lost, assuming a Rankine cycle to be employed between the new pressure limits? Ans. 8 per cent.

6. The mean effective pressure of an actual indicator card is 45 lbs. The theoretical mean effective pressure for the same ratio of expansion, back pressure, and amount of compression, is 48 lbs. What is the card factor of the engine? Ans. 94 per cent.

7. The mean effective pressure obtained from the theoretical indicator card for a locomotive is 126 lbs. What will be probable actual mean effective pressure? Ans. 94 to 106 lbs.

8. An engine cylinder is 10 in. in diameter and the area of the ports is 8 sq. ins. The mean piston speed is 600 ft. per minute. What is the nominal velocity of the steam? Ans. 5,850 ft. per minute.

9. An engine having a cylinder 18 ins. in diameter and a 3-ft. stroke, makes 125 revolutions per minute. Assuming a nominal steam velocity of 5,000 ft. per minute, what will be the area of the exhaust ports? Ans. 38 sq. in.

10. A non-condensing engine takes steam at a pressure of 100 lbs. absolute and has a ratio of expansion of 3. The area of wall surface exposed to the action of the steam per revolution at cut-off is 10 sq. ft. and the number of revolutions per minute is 200. Find the weight of steam condensed per revolution. Ans. 1.0276 lbs.

11. The low pressure piston of a triple expansion engine is provided with poppet valves, so that the area of wall surface exposed to the action of steam at cut-off is that of the cylinder head, piston, and the barrel. The initial pressure is 14 lbs. absolute and the condenser pressure 1 lb. absolute. The cylinder is 80 ins. in diameter and 5 ft. stroke. The engine makes 30 revolutions per minute. The ratio of expansion is 2. Find the weight of steam condensed per revolution in per cent of the cylinder feed per revolution, assuming no clearance volume. Ans. 35 per cent.

12. Construct an indicator card for an engine taking steam at 100 lbs. absolute and discharging it at 16 lbs. absolute with a ratio of expansion of 3, having 10 per cent clearance and complete compression, assuming all compression and expansion lines to be hyperbolic, and that the quality of the steam in the cylinder at cut-off is 50 per cent and at compression 100 per cent. Find the work done per pound of working fluid.

13. Find the work done per pound of steam supplied.

14. Assume the same conditions as before except that there is no compression, and find the work done per pound of working fluid.

15. Find the work done per pound of steam supplied. (Note the effect of compression on the efficiency.)

16. A friction card is taken from an engine and its area found to be 0.16 sq. in. The card at full load of the same length has an area of 1.55 sq. ins. What is the mechanical efficiency of the engine? Ans. 89.7 per cent.

CHAPTER XI

NOTES ON THE DESIGN AND TESTING OF STEAM ENGINES

185. Choice of Type of Engine. In designing a steam engine, it is first necessary to settle upon the type of engine and the range of steam pressure to be employed. The type chosen will depend upon the power required, and upon the use to which the engine is to be put, and is settled primarily by financial and not by purely thermodynamic considerations. It is desirable that the cost of operation of the power plant shall be a minimum. This cost of operation includes three principal elements, the first being the cost of fuel, the second the cost of attendance, and the third the interest and other fixed charges on the first cost of the plant. An engine which is highly economical in the use of fuel usually will be costly, and hence the fixed charges will be large. If an engine is to be operated for a large part of the time, or if fuel is expensive, the cost of the fuel becomes the most important element in the cost of operation, and a highly efficient type of engine will be chosen, in spite of its first cost. If the engine is to be used only a small part of the time, or if it is small in size, or if the fuel is cheap, the fixed charges become the largest item in the cost of operation. In such a case the cheapest engine will be chosen in spite of its low efficiency.

In this connection it should be remembered that it is not the cost of the engine alone, but the cost of the whole plant which we desire to reduce. The more efficient the engine, the smaller the boiler which will be required to operate it. If a cheap engine is very inefficient, the boiler required may become so large and costly that the total cost of the plant will be greater than it would be if a more costly, but more efficient engine had been chosen. Hence, when comparing the cost of operation of two engines, it is necessary to take account of the size and cost of the boiler plant required to operate each of them.

It is not often, however, that the designer of an engine is called upon to fix the type and horse-power of the engine, or the range of steam pressure to be employed. That is usually the work of the consulting engineer who designs the power plant. While large engines are almost always built to order, they are usually built from standard drawings and patterns, which have been prepared in anticipation of the probable requirements of power plant engineers. When a number of such designs have

been submitted to him, together with the prices of the engines, it is the duty of the consulting engineer to choose the particular one which will give the lowest plant operating cost. The type and size of engine which a manufacturer will attempt to build will depend on the facilities of his plant, and the apparent demands of the market. Having originated a series of sizes, he will then, by making minor changes in his designs and patterns, seek to adapt them in the best manner possible to the needs of each particular case, as they are outlined by the consulting engineer.

186. Cylinder Arrangements for Multiple Expansion Engines. The multiple expansion engine is an engine in which the steam performs work in two or more cylinders in succession, in the manner already described in Chapter VIII, Art. 128. Many different cylinder arrangements are used for such engines. In the case of compound engines, the two cylinders may be in line with one another with the two pistons upon a common piston rod as in Fig. 77 *a*. The first cylinder into which the steam enters is called the high pressure cylinder, while the second is called the low pressure cylinder. They are indicated by the letters H.P. and L.P. in the diagram. Rec. is the receiver placed between the cylinders. This arrangement is termed a tandem compound engine. A second arrangement is shown in Fig. 77 *b*, and is known as a cross compound engine. The two cylinders are side by side, each one acting upon a separate crank, which is keyed to a common shaft. In order to obtain a more even turning moment the two cranks are placed at right angles to one another, so that when one of the cylinders is at dead center, the other one will be at mid-stroke. A compound engine may have two L.P. cylinders, in which case it is called a three-cylinder compound. The cylinders are then usually arranged side by side, and act upon three separate cranks set at 120° to each other, all keyed to a common shaft as shown in Fig. 77 *c*. A fourth arrangement is that known as the angle compound engine shown in Fig. 77 *d*, in which one cylinder (usually the H.P.) is horizontal, and the other is vertical. Both act on a common crank pin, and since one cylinder is at dead center while the other is at mid-stroke, the same uniform turning moment is obtained as is gotten from a cross compound engine.

A fifth arrangement is that termed the duplex compound, in which the H.P. and L.P. cylinders both act on a common cross-head, as shown in Fig. 77 *e*. A sixth arrangement, which permits the designer to dispense with the receiver, is termed a Wolff compound, and is illustrated in Fig. 77 *f*. The two cross-heads are linked to opposite ends of a walking-beam, so that the two pistons move in opposite directions. Admission to the L.P. cylinder occurs directly through the exhaust valve of the H.P. cylinder, and continues for almost the entire stroke. Various

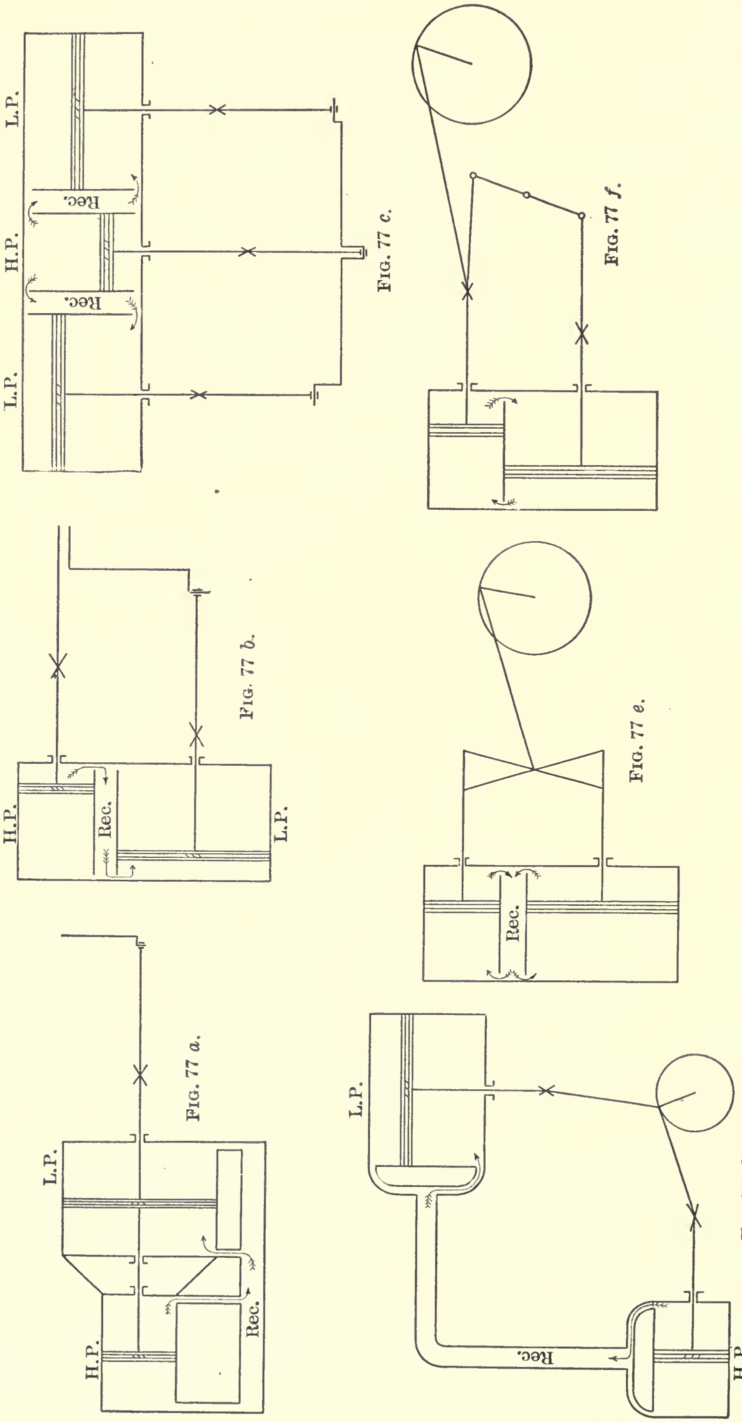


Fig. 77.—Cylinder arrangements for compound engines.

other cylinder arrangements are occasionally used in practice, but the ones given are the most common.

The triple expansion engine usually has three cylinders, termed respectively the high pressure, the intermediate, and the low pressure cylinders. The two receivers are known as the first and second receivers. The three cylinders are usually placed side by side and act on three cranks keyed to a common shaft and placed at angles of 120° with one another, as shown in Fig. 78 *a*. Four cylinder triple expansion engines are often built,

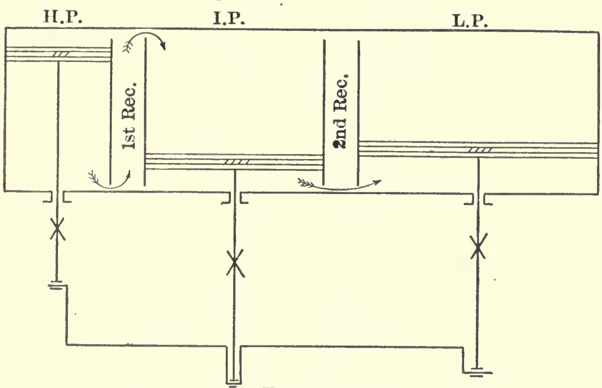


FIG. 78 *a*

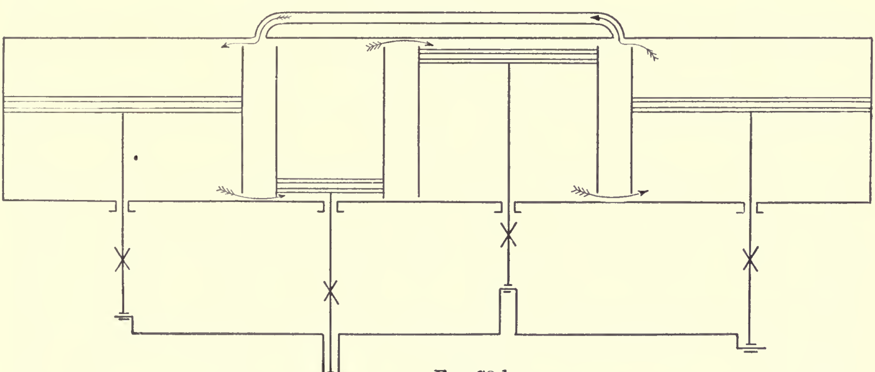


FIG. 78 *b*

FIG. 78.—Cylinder arrangements for triple expansion engines.

having two low pressure cylinders. They are arranged as shown in Fig. 78 *b*. Quadruple expansion engines are seldom built except for merchant marine service, and often have 5 or 6 cylinders, acting on as many cranks keyed to a common shaft. The introduction of the steam turbine which is capable of utilizing low pressure steam to better advantage than the steam engine has tended to minimize the importance of triple and quadruple expansion engines. Greater economy can be obtained by utilizing

the steam from a compound engine in a low pressure steam turbine than by further expanding it in additional cylinders.

187. Advantages of Multiple Expansion. The multiple expansion engine offers several advantages over a single engine having the same ratio of expansion. They are:

First, reduced cylinder condensation, on account of the reduction in the temperature range and ratio of expansion per cylinder.

Second, reduced leakage loss, on account of a reduction in the pressure difference which causes the leakage.

Third, higher mechanical efficiency, since the ratio of the maximum to the mean effective pressure in each of the cylinders is greatly reduced, being usually from 40 to 70 per cent of what it would be were the same total ratio of expansion employed in a single cylinder engine.

Fourth, the principal parts of the engine are less heavy and costly, since the maximum total steam pressure on each of the pistons is only from 20 to 30 per cent of what it would be were a single cylinder engine employed having the same total ratio of expansion.

Fifth, by causing two or more cylinders to operate on separate cranks on the same shaft, as is done in the cross compound engine, a more even turning moment may be secured, which is a matter of very great importance in the case of engines operating alternating current generators in parallel, and is desirable in many other cases.

188. Action of the Steam in a Compound Engine. In order to make clear the action of the steam in the cylinders of a compound engine, the simplest possible case will be considered. Assume a compound engine in which the weight of cushion steam is the same per stroke for each cylinder; the H.P. cylinder is without compression, and the L.P. cylinder has complete compression. The weight of cylinder feed per stroke will of course be the same for each cylinder, since all of the steam leaving the high pressure cylinder must pass through the low pressure cylinder before it is finally discharged from the engine. Assume that the receiver is of very large volume, so that no change of pressure results when the H.P. cylnder is discharged into it or the L.P. cylinder takes steam from it. If there is no loss from wire drawing the pressure of the steam entering the L.P. cylinder will be the same as that exhausted from the H.P. cylinder. If the volume of the steam taken per stroke by the L.P. cylinder is the same as the volume of the steam discharged by the H.P. cylinder, the H.P. cylinder will have complete expansion and the form of its card will be that bounded by the lines $a\ b\ c\ d$ in Fig. 79. The L.P. cylinder will take the same volume of steam from the receiver as was discharged into it by the H.P. cylinder, and its card will be that bounded by the lines $d\ c\ e\ f\ g$ in the same figure. An inspection of the two cards will serve

to show that they are simply the theoretical card of an engine having a large ratio of expansion.

The work done by the steam in passing through the engine is the same as the work which that steam would do in the L.P. cylinder of the engine, if the steam were admitted to the cylinder at boiler pressure, and a sufficiently short cut-off were used to get the same ratio of expansion in the L.P. cylinder as actually occurs in the entire engine. The addition of the high pressure cylinder does not increase the power of the engine, but it does result in gaining the advantages already enumerated in the preceding article. On this account, when the range of steam pressure available is great enough to make a large ratio of expansion desirable, a compound engine is almost always chosen, rather than a simple engine of the same power.

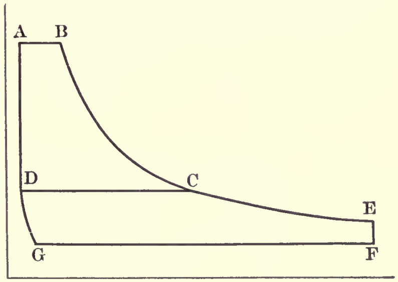


Fig. 79.—Theoretical card for a compound engine.

It is not often, however, that the clearance volumes and points of compression in the high and low pressure cylinders of an engine are so adjusted that the weight of cushion steam contained in each cylinder is the same. Furthermore, the receiver is never of sufficiently large volume to eliminate pressure changes due to the discharge of steam by the H.P. cylinder and draft of steam by the L.P. cylinder. On this account, the usual behavior of the steam in the cylinders of a multiple expansion engine is not quite the simple matter that has been outlined. However, the error introduced by estimating the power of a multiple expansion engine from the area of the theoretical card already described and the volume of its low pressure cylinder, is so small that it may be neglected in designing such an engine. It is customary in design work to fix the size of the low pressure cylinder of a multiple expansion engine by assuming that steam is admitted to that cylinder at boiler pressure, and that the total range of expansion occurs there.

189. Determination of Cylinder Dimensions. When the size and type of engine and range of steam pressure have been settled, either to meet a given set of operating conditions or to meet the probable requirements of the market, it is next in order to determine the probable mean effective pressure and the size of cylinder (or in the case of a multiple expansion engine, the size of L.P. cylinder) required to develop the power. In order to do this, it is usual to lay out to scale the theoretical card for the pressure range and ratio of expansion which it has been decided to adopt. Having drawn the theoretical card, the designer

measures or calculates its area and from its area and length the mean ordinate of the card is obtained. Multiplying together the height of the mean ordinate, and the pressure scale of the drawing, the theoretical mean effective pressure is obtained. The theoretical mean effective pressure should then be multiplied by the proper card factor in order to obtain the actual mean effective pressure at the rated load of the engine.

The area of the piston of the engine is now obtained by the formula

$$A = 33{,}000\ \frac{HP}{PS},$$

in which A is the area of the piston in square inches, HP is the indicated horse-power of the engine at rated load, S is the mean piston speed in feet per minute, and P is the mean effective pressure in pounds per square inch. The mean piston speed is of course equal to twice the length of the stroke in feet times the number of revolutions per minute. The area so obtained will of course, be the area of the low pressure piston in the case of a multiple expansion engine. In case the engine has two or more L.P. cylinders, it is the combined area of all the L.P. pistons.

If the engine is to be a compound or triple expansion engine, the theoretical card is now divided by horizontal lines into two or three portions as nearly equal in area as may be. This is done in order that the amount of work developed in each of the cylinders shall be the same. Fig. 80 represents such a card as would be laid out in designing a compound engine. The horizontal line $e\ f$ divides the card into two portions of nearly equal area. The high pressure card which is the upper of the two areas is now modified by giving a certain amount of terminal drop as shown by the line $c\ d$. The reasons for giving this terminal drop are that it reduces the size of the high pressure cylinder, increases the mechanical efficiency of the engine, and reduces the loss due to cylinder condensation. The length $e\ d$ will now represent the swept volume of the H.P. cylinder while the length $i\ h$ will represent the swept volume of the low pressure cylinder. If both cylinders are of the same length of stroke, the ratio of $e\ d$ to $i\ h$ also represents the ratio of the piston areas of the two cylinders. Having determined the area of the low pressure piston by the rule already given, the area of the high pressure piston may be determined by this ratio. The length of stroke may next be chosen,

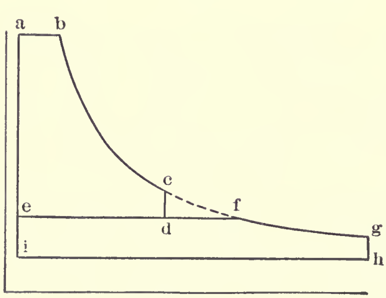


Fig. 80.—Card used in designing a compound engine.

and the number of revolutions per minute be determined by twice the length of stroke in feet.

190. The Design of Receivers. In order that the low pressure cylinder may receive its supply of steam without too much variation in pressure, and that the high pressure cylinder may exhaust its steam, while the low pressure inlet valves are closed, a receiver of considerable volume is interposed between the high and low pressure cylinders. The volume of the receiver is usually made from 2 to 6 times the volume of the high pressure cylinder. The larger the volume of this receiver, the higher the card factor of the engine, and the less the loss due to overlapping of the cards. However, if the receiver is made too large, the loss due to radiation will overcome the advantage of an improved card factor, so that the limits given represented the practical range in variation of receiver volume. In the case of a triple expansion engine, the volume of the second receiver, interposed between the intermediate and low pressure cylinders, is from $1\frac{1}{2}$ to 4 times that of the intermediate cylinders. It may be shown that the relative position of the cranks of an engine has an important effect on the range of pressure variation in the receiver. The cranks should be so placed that this pressure variation will be a minimum.

On account of its adiabatic expansion and the loss of heat by radiation, the steam which enters the receiver will be wet. If this wet steam is permitted to enter the low pressure cylinder, the cylinder condensation will be greatly increased on account of the excessive wetness of the steam. To avoid this it is customary to heat the steam in the receiver by means of a coil of pipe called a reheater. The reheater is supplied with steam of a higher pressure than that in the receiver, and on account of its high temperature it evaporates the water, thus supplying the second cylinder with dry steam. However, this action is accompanied by a thermodynamic loss and is practically the equivalent of operating the engine on a jacketed cycle. A preferable method is to so form the receiver that it becones a separator, mechanically removing the moisture contained in the entering steam, and supplying the second cylinder with steam that is practically dry. A reheater may be used to advantage in connection with a separating receiver, but the amount of heat supplied by the reheater will then be very small. The use of the reheater will give almost absolutely dry steam to the following cylinder, with a resulting decrease in the loss from cylinder condensation.

The amount of reheater surface used varies greatly in different types of engines, and is dependent very largely on the judgment of the designer. Good practice sanctions the use of from 0.02 to 0.05 square feet of reheating surface per pound of steam per hour. In case a separating receiver is used, the area of reheating surface may, of course, be greatly diminished.

191. Design of Jackets. In the case of very slow-moving engines, as for instance long stroke pumping engines, the cylinders should be jacketed. It is customary, when practicable, to jacket both the barrel and the heads of the cylinder. It is equally advisable to jacket the piston although this is seldom attempted on account of the difficulty of introducing steam and carrying away the drip through the piston rod. When jacketing an engine, care must be taken that the jackets are drained, so that water will not accumulate in them, since a jacket filled with water tends to increase rather than diminish the loss due to cylinder condensation. It is not difficult to drain the jackets of the cylinder barrel, but ofttimes considerable ingenuity must be used in order to drain the jackets of the heads, when the valves are placed in the heads, as they are usually in the case of slow speed vertical engines. The reheating coils in the receivers must also be so arranged that they can be drained. The jacket drain should lead to a trap which will discharge the accumulated water without permitting the escape of steam. In the case of a high pressure cylinder, this trap should discharge into the receiver, as should also the drip from the reheating coil, in order that the sensible heat of the water may be utilized in evaporating a portion of its weight into steam, which will do work in the low pressure cylinder. In a triple expansion engine, the drips from the intermediate cylinder and the reheater coil of the second receiver in like manner may discharge into the second receiver.

When a highly efficient multiple expansion engine is desired, an approximation to the Carnot cycle may be obtained by pumping the feed-water from the hot well (i.e., the chamber into which the air-pump discharges the condensed steam) through heating coils surrounded by steam from the receivers. The steam used to heat the feed-water has already done work in one or more cylinders, and the feed-water is finally supplied to the boiler at practically the temperature of the steam in the first receiver. While this method makes an engine highly economical, the economy of the plant will be lower than if an economizer were used. Although this method has been employed in practice, it was used, not in order to secure a high plant economy, but in order to earn a bonus for a high engine economy.

Having settled upon the areas of the pistons, the length of stroke, the number of revolutions per minute of the engine, the volume of the receivers and the amount of reheating surface, the areas of the ports may be computed by the rules given in Art. 170. The remainder of the design of the engine becomes a problem in machine design in which the principles of thermodynamics play no part.

192. Theoretical Indicator Cards for Multiple Expansion Engines. In order to construct accurately the theoretical indicator cards for a multiple expansion engine, it is necessary to take account of the volume of the receivers, and the pressure

variations which take place within them. It is customary to compute the form of the cards for such engines on the assumption that the product of the pressure and the volume of a given weight of steam is a constant quantity. This is not exactly true, but is a sufficient approximation in estimating the power and proportioning the cylinders of such engines. Having designed the cylinders and the receivers of a multiple expansion engine, and, from the design, computed the clearance volume of each of the cylinders, we are in position to draw the theoretical indicator cards of the several cylinders. In order to illustrate the method of constructing such cards, a cross compound engine will be assumed.

In such an engine steam is admitted at boiler pressure to the H.P. cylinder up to the point of cut-off. After the H.P. inlet valve is closed, the steam in that cylinder expands until the end of the stroke, when the exhaust valve opens. If, as is usually the case, the receiver pressure is less than the terminal pressure of the H.P. cylinder, there will be a sudden drop in pressure in the H.P. cylinder, and a sudden rise in the pressure in the receiver, at the instant of release. The H.P. piston will now begin to return, compressing the exhaust steam into the receiver and consequently raising the pressure, both in the receiver and in the H.P. cylinder. When the H.P. piston reaches mid-stroke, the L.P. piston arrives at the end of its stroke and the L.P. inlet valve opens. Unless compression is complete in the L.P. cylinder, steam will rush into this cylinder from the receiver, and the receiver pressure will drop. During the remainder of the back stroke of the H.P. piston, the L.P. piston is moving forward, and the L.P. cylinder is taking steam at a faster rate than it is discharged by the H.P. cylinder. Since the steam is increasing in volume, its pressure will fall until the L.P. inlet valve closes. If this occurs before the H.P. exhaust valve closes, the pressure in the receiver and the H.P. cylinder will again begin to rise. If it occurs after the H.P. exhaust valve closes, the pressure in the receiver will not rise and the pressure in the H.P. cylinder will rise only an account of compression.

Fig. 81 is an illustration of the theoretical card given by such an engine; *a–b* is the steam line of the H.P. cylinder; *b–c* is the expansion line of the H.P. cylinder; *c–d* is the terminal drop of this cylinder; *d–e* is the period during which the H.P. exhaust is being compressed into the receiver before the L.P. inlet valve opens; *e–f* represents the drop in pressure in the H.P. cylinder when the L.P. inlet valve opens; *f–g* represents the period during which the H.P. exhaust and the L.P. inlet valves are both open; *g–h* is the compression period in the H.P. cylinder; *f′–g′* is the admission period of the L.P. cylinder up to the time when the H.P. exhaust valve closes; *g′–i′* is the remainder of the admission period, which occurs while the H.P. exhaust is closed; *i′–j′* is the expansion period of the L.P. cylinder, *j′–k′* represents the L.P. terminal drop, *k′–l′* the L.P. exhaust; *l′–e′* represents the L.P. compression period, and *e′–f′* represents the rise in pressure in the L.P. cylinder at the point of admission which is coincident with the fall in pressure, *e–f*, in the H.P. cylinder and the receiver.

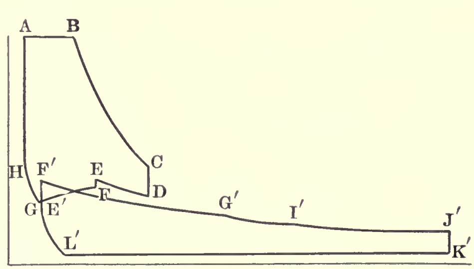


FIG. 81.—Computed card for a cross compound engine.

In computing the card for such an engine, it is necessary to know the pressure and volume at L.P. release, ($j′$), and the initial and back pressure (i.e., at *ab*, and *k′–l′*).

It is also necessary to know the receiver volume (which we will designate by the symbol R) and the volumes of the high and low pressure cylinders at points h, b, c, e, g, e', g', i', j', and l'. These may be determined graphically when the clearance volumes, and the points of cut off and compression are known for each cylinder. In order to illustrate the method of computing the card the following example will be assumed: The initial steam pressure is 150 pounds, the L.P. terminal pressure 10 pounds, and the back pressure 2 pounds per square inch absolute. The swept volume of the H.P. cylinder is 4 cubic feet and of the L.P. cylinder 15 cubic feet. The H.P. clearance volume is 0.4 cubic feet (i.e., 10 per cent) and the L.P. clearance volume 0.6 cubic feet (i.e., 4 per cent). The receiver volume is 12 cubic feet. The point of compression is 10 per cent from the end of the stroke for each cylinder, and L.P. cut-off occurs at 40 per cent of the stroke. We will now have

$$V_j' = 15.6 \text{ cu.ft.}$$

$$V_i' = 6.6 \quad \text{"}$$

$$V_e' = 2.1 \quad \text{"}$$

$$V_a = .4 \quad \text{"}$$

$$V_c = 4.4 \quad \text{"}$$

$$V_e = 2.4 \quad \text{"}$$

$$V_g = .8 \quad \text{"}$$

In order to find the volume of the L.P. cylinder at point g', we may make the construction shown in Fig. 82, in which points H and L represent the simultaneous positions of the H.P. and the L.P. cranks at the point of compression in the H.P. cylinder, distance $a\,b$ represents the distance of the point of compression from the end of the stroke of the H.P. cylinder, and $a\,c$ the distance of the point g' from the end of the stroke of the L.P. cylinder. From such a construction we find the distance $a\,c$ to be 20 per cent of the stroke, and the volume

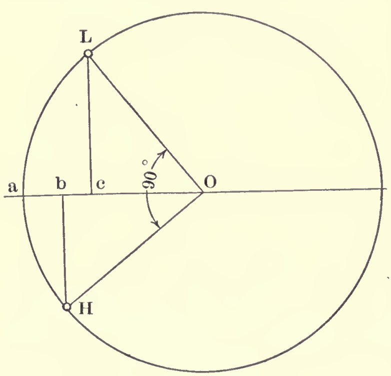


FIG. 82.

$$V_g' = 3.6 \text{ cu.ft.}$$

The product of the pressure and volume of the steam in the L.P. cylinder at release is

$$V_j'\,P_j' = 15.6 \times 10 = 156.$$

The product of the pressure and volume at cut-off is the same, and the pressure

$$P_i' = \frac{156}{6.6} = 23.7 \text{ lbs.}$$

The pressure in the receiver at this point is the same. Consequently the product of the pressure and volume of the steam in the L.P. cylinder and receiver together at this point is

$$156 + 23.7 \times 12 = 440.$$

The volume of the steam in the receiver and L.P. cylinder at point g' is

$$12+3.6=15.6 \text{ cu.ft.}$$

The pressure

$$P_{g'}=P_g=\frac{440}{15.6}=28.2 \text{ lbs.}$$

The product of the pressure and volume of the steam in the H.P. cylinder, the receiver, and the L.P. cylinder at the instant H.P. compression begins is therefore

$$440+28.2\times0.8=462.6.$$

The volume of the steam contained in the H.P. cylinder, the receiver, and the L.P. cylinder at L.P. admission is

$$2.4+12+0.6=15 \text{ cu.ft.}$$

We will therefore have for the pressure

$$P_{f'}=P_f=\frac{462.6}{15}=30.8 \text{ lbs.}$$

The product of the pressure and volume of the steam contained in the L.P. cylinder during compression is

$$2\times2.1=4.2.$$

The product of the pressure and volume of the steam in the low pressure cylinder at point f' is

$$30.8\times0.6=18.5.$$

The product of the pressure and volume of the steam in the H.P. cylinder and the receiver at point e is therefore

$$462.6-4.2=458.4.$$

The product at point f is

$$462.6-18.5=444.1.$$

Since the volume of the steam at both points is $12+2.4=1.44$ cu.ft., the pressure at e is

$$P_e=\frac{458.4}{14.4}=31.8 \text{ lbs.}$$

The volume of this steam at d is

$$12+4.4=16.4 \text{ cu.ft.,}$$

and its pressure is

$$P_d=\frac{458.4}{16.4}=27.9 \text{ lbs.}$$

The pressure of the steam in the receiver at H.P. release is $P_i'=23.7$, and the product of its pressure and volume is therefore

$$23.7\times12=284.$$

The product of the pressure and volume of the steam in the H.P. cylinder at release is therefore

$$458.4-284=174.4.$$

The pressure at release was therefore

$$P_c=\frac{174.4}{4.4}=39.7 \text{ lbs.}$$

The volume at H.P. cut-off was

$$V_b=\frac{174.4}{150}=1.162 \text{ cu.ft.}$$

and cut-off occurs at 19 per cent of the stroke.

The pressure at e' is

$$P_{e'}=\frac{4.2}{0.6}=7 \text{ lbs.}$$

the pressure at h is

$$P_h=\frac{P_g\times V_g}{V_h}=\frac{28.2\times 0.8}{0.4}=56.4 \text{ lbs}$$

The volume of the cylinder feed is

$$\frac{174.4-28.2\times 8}{150}=1.01 \text{ cu.ft.}$$

The indicated steam consumption per stroke is

$$1.01\times 0.338=0.342 \text{ lbs.}$$

The actual steam consumption per stroke may be estimated by adding to the indicated steam consumption the estimated weight of the steam condensed per stroke, and an allowance for leakage. The power of the engine may be estimated by finding the power of each cylinder shown by the cards obtained, after making proper reduction for wire drawing and steam friction.

193. Combined Cards for Multi-Cylinder Engines. The combined card of a steam engine is obtained by placing together the H.P. card and L.P. card, using the same scale of pressures and the same scale of volumes and setting off the admission line of the two cards at such distance from the zero volume line is as indicated by the clearance of the respective cylinders. As has already been shown, when the same weight of cushion steam is contained in each cylinder, and the compression in the L.P. cylinder is complete and there is no compression in the H.P. cylinder, the combined card (except for overlapping wire drawing, and H.P. terminal drop) is the same as would be given in a single cylinder engine having the volume of the L.P. cylinder and the same total ratio of expansion. If the same weight of cushion steam is contained in each cylinder, but the conditions of compression are different from those given, the card will be that which would be given by a cylinder of different swept volume, as may be seen in Fig. 83, in which the length A

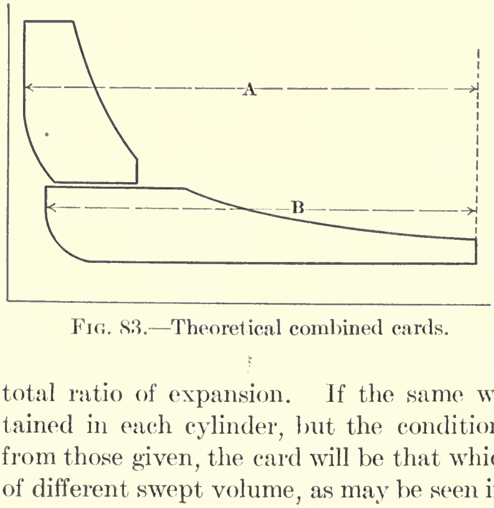


Fig. 83.—Theoretical combined cards.

represents the swept volume of the theoretical cylinder required to develop the power shown by the card, and length B represents the swept volume of the actual low pressure cylinder. The swept volume plus the clearance volume for the two cylinders, however, will be the same.

When the weight of cushion steam contained in the two cylinders is different, it will be impossible to draw correctly a combined card which will represent the action of the steam, since the weight of steam represented by the H.P. card will be different from that represented by the L.P. card. In such cases, it is customary to refer the two cards to the same pressure and volume scales. The theoretical expansion line for the two cards will not, however, be the same.

194. Testing Steam Engines. In making a test of a steam engine it is usual to obtain the following data:

First, the pressure of the steam supplied to the engine.

Second, the quality of steam supplied to the engine.

Third, the weight of wet steam rejected by the engine.

Fourth, the pressure of the steam in the condenser.

Fifth, the temperature of the water entering and leaving the condenser.

Sixth, the weight of the drip from each of the jackets.

Seventh, the number of revolutions per minute made by the engine.

Eighth, indicator cards are taken from each end of each cylinder.

Ninth, the brake horse-power of the engine is obtained.

The precautions which must be observed in making such a test to insure that these data are properly taken have been outlined by a committee of the American Society of Mechanical Engineers, in their standard methods of testing steam engines.[1] The method of testing here outlined involves the use of a surface condenser. When any other type of condenser is employed, the weight of water fed to the boiler and not the weight of steam rejected by the engine, must be taken as the measure of the steam supplied. An engine test should last several hours and the conditions of load, steam pressure, vacuum, etc., should be kept as nearly constant as possible. Any considerable variation in these quantities during the test will invalidate the results. The readings are taken at frequent intervals, usually every ten minutes.

195. Graphical Analysis of an Engine Test. It is instructive in working up the results of an engine test to superimpose the indicator cards of the engine upon the theoretical diagram of the cycle in order to determine the magnitude and distribution of the losses which occur. In order to do this, it is necessary to construct a mean card for each of the cylinders, which will represent the average conditions for both the head and crank ends of that cylinder for the entire test.

After computing the average mean effective pressure developed during the entire

[1] See the Transactions of the A.S.M.E. for 1902. The rules are also published by the Society in pamphlet form.

test in the head and crank end of each cylinder, that set of indicator cards is chosen in which the mean effective pressures are nearest to the average. These cards are, f course, the best representative cards for the test. Each of the cards may then be ruled with a number of equidistant vertical lines, as shown in Fig. 84. Upon the paper on which the mean card is to be constructed for any cylinder, rule a horizontal line, for the atmospheric line, making its length represent the swept volume of both ends of the cylinder, to any suitable scale. Upon it erect the same number of equidistant vertical lines as has already been drawn upon each of the indicator cards.

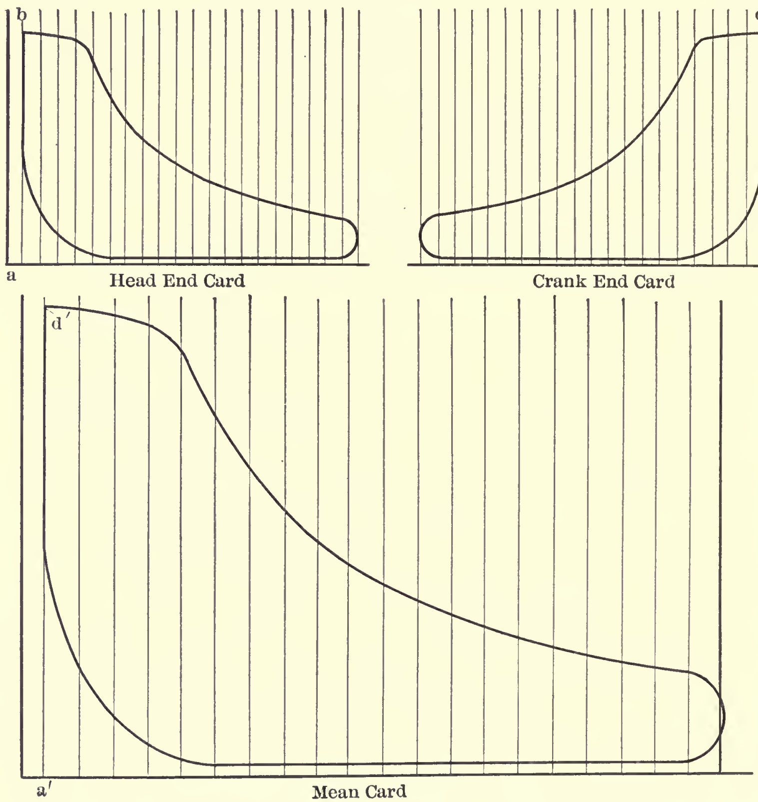


FIG. 84.—Construction of a mean card.

Upon the first vertical line of the head end card, measure the distance from the atmospheric line to a point on the outline of the card, as *a–b*. Add to this distance the corresponding distance *c–d* measured upon the crank end card, and lay off upon the first vertical line of the mean card the sum of the distances *a–b* and *c–d*, at *a′d′*. In like manner lay off all the other points and so draw a card which is the mean of the head and crank end cards for the cylinder. The zero pressure line may now be drawn parallel with the atmospheric line, the distance between the lines being determined by the atmospheric pressure at the time of the test, as computed from the barometer reading. The zero volume line is drawn perpendicularly to the atmospheric line at such a distance from the end of the mean card as represents the sum of the head and

crank end clearance volumes. The theoretical indicator card for a Rankine cycle with complete compression is now constructed, the weight of cushion steam and the weight of working fluid being the same for the Rankine cycle as it is for the cylinder in question. The quality of the steam at cut-off and at the beginning of compression, in the Rankine cycle, is assumed to be the same as that of the cylinder feed, as determined during the engine test. The pressure limits of the Rankine cycle are, in the case of the H.P. cylinder of a multiple expansion engine, the pressure of the steam at the throttle valve and that of the steam in the receiver. In the case of an intermediate cylinder, the pressure limits of the Rankine cycle are the pressures of the steam in the preceding and following receivers. In the case of the L.P. cylinder of a multiple expansion engine, they are the pressure of the steam in the preceding receiver and the pres-

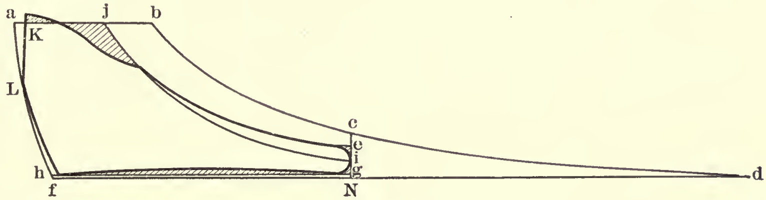


FIG. 85.—Actual card superimposed upon a Rankine cycle card to show the losses.

sure corresponding to the temperature of the discharged condensing water. In the case of a simple non-condensing engine, the pressure limits are the pressure of the steam at the throttle and the pressure of the atmosphere.

Fig. 85 shows the construction for the L.P. cylinder of a compound engine, the heavy outline being the actual indicator card, while the light outline is the theoretical card for the Rankine cycle. Since terminal drop is employed, the toe of the Rankine cycle, bounded by the lines *cdN*, will be lost. This loss is due to the imperfection of the cycle employed. A line *g–h* drawn parallel with the line *f–d* (which represents the pressure corresponding to the temperature of the discharged condensing water) and tangent to the actual card, marks off the area *ghfN* is which represents the loss due to the imperfection of the condensing apparatus. The expansion line

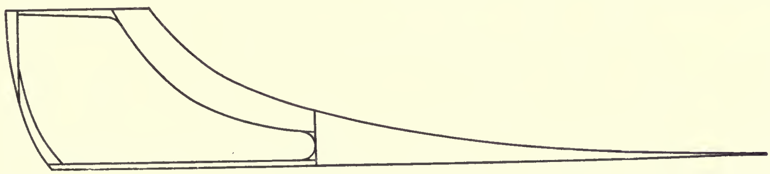


FIG. 86.—Card from a non-condensing engine superimposed on a Rankine cycle card.

of the steam may now be completed, and is represented by the line *je*. The area *j–b–c–e* will then represent the loss of power due to cylinder condensation. The shaded area minus the dotted area represents the loss of power due to the combined effects of steam friction, wire drawing, and overlapping. The smaller the volume of the receiver, the larger will be the amount of this loss. The area *a–K–L* represents the loss due to clearance. There is a further loss due to clearance included in the area *c–d–f* on account of the incomplete expansion of the steam compressed in the clearance spaces. If an adiabatic expansion line *j–i* be constructed from point *j*, the area *e–j–i* will be the work restored by the re-evaporation of the steam initially condensed. A part of this work so restored is, of course, lost from other causes. The area in the lower left-

hand corner, bounded by the compression line and by the line $L–h$, is the work lost on account of the compression of dry and saturated steam which results from the heating of the exhaust by the cylinder walls.

Fig. 86 represents a card similarly treated for a simple non-condensing engine. The losses may be traced out by the reader.

196. Other Methods of Analyzing Engine Tests. A method formerly much in use in analyzing the results of an engine test is that known as Hirn's analysis. In this method, the heat transferred to or from the steam contained in the cylinder is determined for each portion of the cycle. It is assumed in Hirn's analysis that all heat transfers not otherwise accounted for are due to cylinder condensation or re-evaporation. This is not true, since such heat transfers are often due to leakage. The computations involved in Hirn's analysis are rather laborious, as may be seen by reference to Art. 55 of the second volume of Zeuner's Technical Thermodynamics, in which the theory of Hirn's analysis is fully developed. Hirn's method of analysis emphasizes unduly the losses due to cylinder condensation, and does not separate or analyze other sources of loss. On this account, Hirn's method of analysis is not much used at the present time in determining the amount and distribution of the losses in the steam engine.

Another method of determining the losses in a steam engine is to draw the temperature-entropy diagram of the fluid contained in the cylinder and then to superimpose this diagram upon the theoretical temperature-entropy diagram of the cycle in the manner shown in Chapter XXV. The temperature-entropy diagram has the advantage of illustrating more clearly the heat transfer to and from the cylinder walls, but it is more difficult to employ than the methods described in Art. 195, which are, on the whole, the most satisfactory methods of analyzing the results of an engine test.

197. Methods of Comparing Engine Efficiencies. Many methods are in use for stating the efficiencies of steam engines. One of the commonest is to determine the number of pounds of dry steam supplied per hour to the engine per indicated horse-power developed. This is usually known as the **water rate** of the engine. While, in general, a low water rate means a highly efficient engine, the water rates of different types of engines are not proportional to their true economy. Accordingly, a second method has been suggested in which the efficiency of the engine is expressed in terms of its heat rate (i.e., in terms of the number of B.T.U. supplied to the engine per indicated horse-power per hour). The heat rate may be derived from the water rate by subtracting from the total heat of the dry steam supplied, the heat of the liquid at the temperature of exhaust, and multiplying by the water rate. Sometimes the number of B.T.U. supplied per brake horse-power is given. In the case of direct connected units, the efficiency of the combined unit is often expressed by the number of B.T.U. supplied per kilowatt hour output of the generator. Occasionally the number of B.T.U. per minute is made the basis of the statement of efficiency. The efficiency is often expressed as a per cent of the theoretical efficiency of the Rankine cycle. The efficiency referred to the Rankine cycle may be obtained by dividing the

heat rate of the Rankine cycle for the given temperature range by the heat rate of the actual engine for the same temperature range. The total efficiency of an engine is sometimes given; which is the per cent of the total heat supplied which is actually transformed into useful work. In the table given in Art. 210, will be found recent figures for the best efficiencies of steam engines and turbines, in which the efficiencies are stated in the different ways most commonly employed.

198. Effect of Variation in the Load on the Efficiency. The power developed by an engine is not a fixed quantity, but is usually automatically varied by the governor in such a manner as to keep the speed of the engine

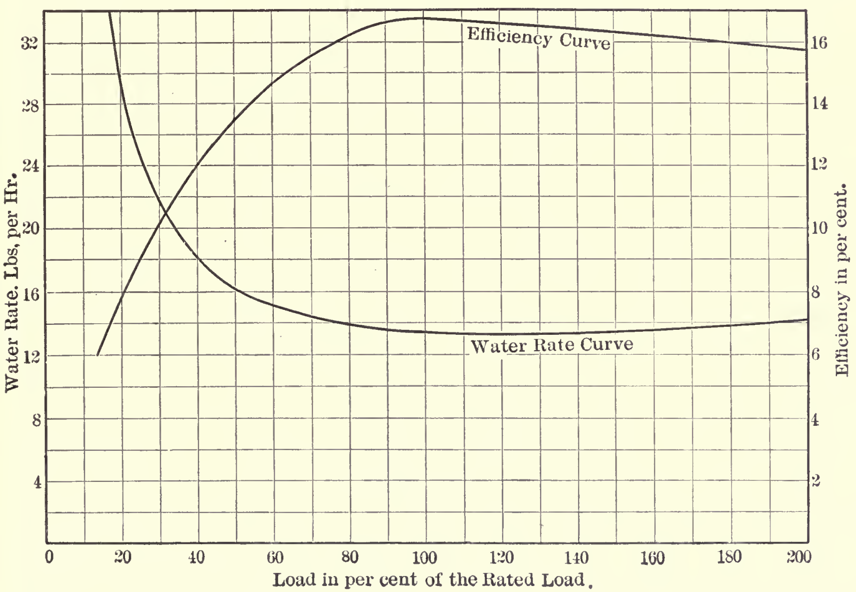


Fig. 87.—Load curve of a steam engine.

constant. The governor accomplishes this by increasing or diminishing the quantity of steam per stroke taken by the engine. As a result of the action of the governor, the form of the steam cycle and the conditions of operation vary as the load on the engine is varied. In consequence of these changes the losses vary, being usually less at low loads than at high loads. If the sum of the losses were always proportional to the power developed, the efficiency of the engine would be constant at all loads. Since such is not the case, it follows that at some particular load the heat rate of the engine will be a minimum (i.e., its efficiency will be a maximum) and at all other loads the heat rate will be increased.

The rated power (i.e. the nominal horse-power) of a steam engine is usually the indicated horse-power at which it gives the minimum heat

consumption per brake horse-power per hour. In the case of other types of heat engines (e.g., steam turbines and gas engines) this is not true, since such engines give the greatest economy at the maximum possible load, and if rated at their most economical load they would have no overload capacity. The efficiency of a steam engine falls off rapidly at low loads, as may be seen by referring to the "load curve" shown in Fig. 87, and it is therefore desirable to operate such an engine at or above its rated load. Consequently when a plant is to furnish a variable amount of power, several engines should be installed, and such a number of them should be operated at any time as will make the load on each one as nearly as possible equal to its rated load.

Furthermore, in comparing the efficiency of different engines, it is necessary to compare their efficiencies at their most economical loads, since comparison on any other basis would be misleading. A statement of the results of an engine test should therefore always include a statement of the actual and of the rated load of the engine, in order that it may be known whether the conditions of operation were such as to make reasonable economy possible.

In the case of an engine with a throttling governor, the total steam consumption of the engine is given approximately by the formula

$$S = A + K \times \text{H.P.},$$

in which S is the total steam consumption, A and K are constants, and H.P. is the brake horse-power developed by the engine. This is known as Willan's Law. It does not hold for cut-off governed engines, for which we may write an approximate formula of the form

$$S = A + K(\text{H.P.})^n,$$

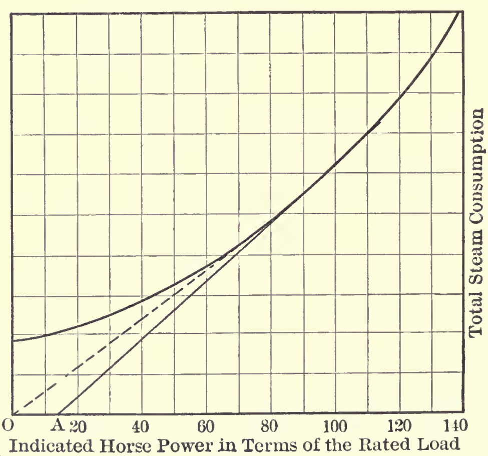


Fig. 88.—Curve of total steam consumptign.

in which n is greater than one. The general form of the curve of total steam consumption for such an engine is shown in Fig. 88. In this figure the segment $O\ A$ is equal to the friction horse-power. A line through O tangent to the curve of total steam consumption will obviously touch it at the point where the water rate per indicated horse-power is a minimum. In like manner, a tangent through A will touch it at the point where the water rate per brake horse-power is a minimum.

A flat load curve is a desirable characteristic in an engine, and when two engines of equal efficiency are compared, that one is the better which has the flatter load curve.

PROBLEMS

1. A steam turbine plant costs $60.00 per kilowatt, while a steam engine plant costs $80.00 per kilowatt. If the fixed charges are 15 per cent per annum in each case, and the plant operates 15 hours per day, for 300 days per year, what will be the costs per kilowatt hour due to fixed charges on the plant?

Ans. 0.200 cents and 0.267 cents.

2. The steam turbine plant requires 2 lbs. of coal per kilowatt hour, and the steam engine plant 1.8 lbs. per kilowatt hour. Coal costs $2.00 per ton. What is the cost per kilowatt hour in each case? Ans. 0.10 cents and 0.09 cents.

3. Which of the two plants will operate at the least total cost per kilowatt hour, disregarding all other costs except those given?

Ans. The turbine plant will operate at 0.300 cents and the engine plant at 0.357 cents per kilowatt hour.

4. Construct a combined card for a compound engine taking steam at 150 lbs. gage and discharging it at 2 lbs. absolute. The clearance of the low pressure cylinder is 5 per cent and the pressure at the end of compression 5 lbs. absolute. The ratio of expansion is 16. Assume hyperbolic expansion and compression.

5. Divide the above card into two parts so that the areas of the two parts are equal.

6. Give sufficient terminal drop to the H.P. card so that the total steam load on the H.P. piston will be equal to that on L.P. piston, at the instant when the load is a maximum in each cylinder. (The total steam load is equal to the area of the piston times the difference in steam pressure at inlet and exhaust.) Assume that the lengths of stroke are equal for the two cylinders and that the areas of the cylinders are proportional to the volumes.

7. Find the mean effective pressure of the above card referred to the L.P. cylinder, assuming a card factor of 90 per cent.

8. What must be the diameter of the L.P. cylinder of an engine, in order that it shall develop 500 indicated horse-power at 600 ft. per minute piston speed, with the M.E.P. found in Problem 7.

9. Find the proper volume for the receiver for the above engine, assuming that it is to be a cross compound engine.

10. An engine uses 10,000 lbs. of steam of 98 per cent quality in a 2-hour test. The indicated horse-power is 250. What is the water rate of the engine?

Ans. 19.6 lbs. per hour.

11. If the steam is supplied at 125 lbs. gage pressure and the condenser pressure is 3 lbs. absolute, what is the heat rate of the above engine? Ans. 21,310 B.T.U.

12. If the mechanical efficiency of the engine is 92 per cent, what is the heat rate per brake horse-power per hour? Ans. 22070 B.T.U.

13. What is the heat rate of the Rankine cycle for the same temperature range?

Ans. 10,810 B.T.U.

14. What is the brake efficiency of the above engine expressed as a per cent of the efficiency of the Rankine cycle? Ans. 49%

15. What is the total efficiency of the above engine? Ans. 11.5%

CHAPTER XII

THE STEAM TURBINE

199. Impulse and Reaction Turbines. The steam turbine is a heat engine which makes use either of the impulse or of the reaction of a jet of steam, in order to transform the energy of this steam into work. If the turbine operates by utilizing the impact of the steam jet, it is known as an **impulse turbine.** If it makes use of the reaction of the steam jet, it is known as a **reaction turbine.** Turbines which combine both principles are sometimes used and are known as **impulse-reaction turbines.** The impulse turbine is sometimes termed the **velocity turbine,** and the reaction turbine is sometimes termed the **pressure turbine.**

200. The Theory of the Turbine Nozzle. If steam be supplied under pressure to a properly shaped nozzle, it will flow from the nozzle with a very high velocity in the form of a jet. If this jet be permitted to strike upon a suitably formed surface so that its direction of motion is changed, as in Fig. 89, the impact of the jet upon the surface will tend to force the surface backwards, and if the surface be permitted to move, work will be performed. The Kerr turbine, illustrated, in Fig. 90, is of this type. If the nozzle itself be permitted to move, the reaction of the escaping steam will force it backward, and work will be performed. This is the principle of the reaction turbine. The Avery turbine, illustrated in Fig. 91, is of this type. It will be seen that the proper operation of a steam turbine will depend upon the form of the nozzles used. It is therefore a matter of primary importance in steam turbine design to make the nozzles of the proper form and size for the work which they are to do. The following paragraphs will serve to make

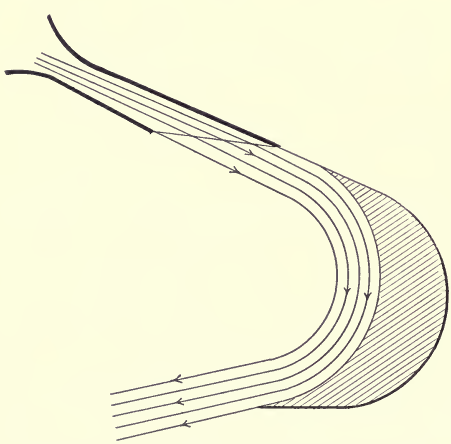


Fig. 89.—Impact of steam jet upon a properly formed vane surface.

clear the action of turbine nozzles and the methods of designing them.

When steam flows through a nozzle each particle will be found to expand in volume and increase in velocity as it passes from the region of high pressure to that of low pressure. In passing through the nozzle, the steam will neither gain nor lose heat. This being the case, the kinetic energy of each pound of steam as it passes a given cross-section of the nozzle, plus the work done by this steam in displacing the steam in the region into which it rushes, must be equal to the loss of internal energy of this pound of steam, plus the work done upon it by the advancing

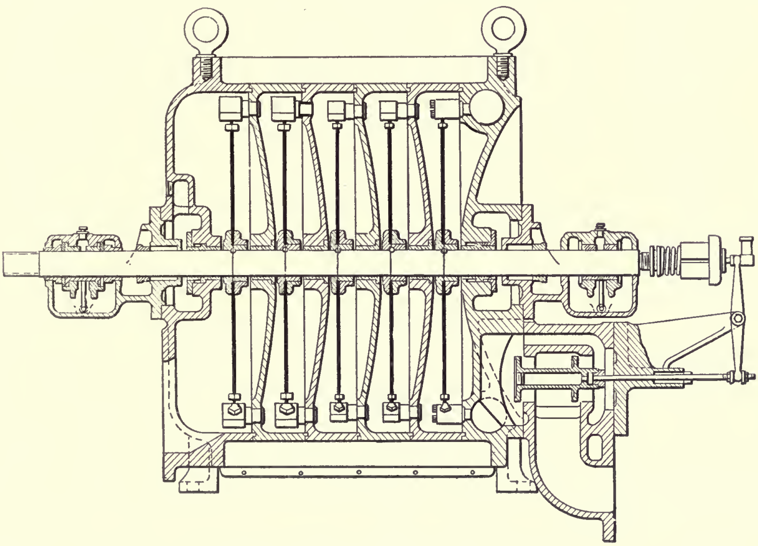


FIG. 90.—Section of a Kerr turbine.

mass of steam which takes its place in the region from which it flows. A consideration of Fig. 92 will make this apparent.

In the figure A is a cylinder and B a nozzle. The cross-section of the nozzle is very small in comparison with that of the cylinder, so that the velocity of the steam in the cylinder may be neglected. The steam flowing from the nozzle passes into the tube C, whose cross-section is the same as that of the nozzle at the point where the nozzle terminates. Assume that cylinder A is filled to the point d with a steam having a pressure P_1 and entropy N, and that tube C is filled to the point E with steam having a pressure P_2. Since the steam neither gains nor loses heat in passing through a frictionless nozzle, the entropy in tube C will be the same as in cylinder A, and the expansion is adiabatic. At E in the tube and at D in the cylinder are pistons which exert upon the

steam a pressure equal to that exerted upon them by the steam. The pressure in tube C being less than that in cylinder A, the steam will flow from A to C through the nozzle, and if the pressures are to remain constant, the pistons must both move to the right. Assume that the proportions of the nozzle are such that 1 pound of steam flows per second, then the work done upon the steam per second by piston D will be the external work of evaporation of 1 pound of steam of pressure P_1 and entropy N. The work done by the steam upon piston E will be equal to the external work of evaporation of 1 pound of steam at pressure P_2 and of entropy N. The kinetic energy of the pound of steam flowing in the tube C will then be equal to the work of expansion (which is the difference between the internal energy of a pound of steam when in cylinder A and in the tube C) plus the work done by piston D, minus the work done upon piston E. This is of course equal to the difference between the total heat of the pound of steam at pressure P_1 and entropy N, and its total heat at the same entropy and at the pressure P_2, a quantity which we will designate by the symbol ΔH, and which is usually termed the **heat drop**. The kinetic energy of a body having a mass of 1 pound is of course

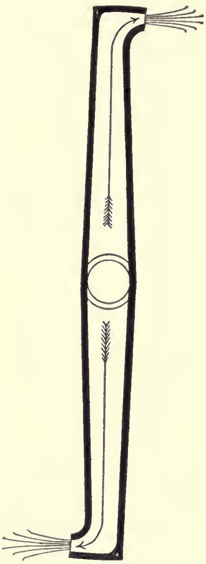


Fig. 91.—Diagram illustrating the principle of the Avery turbine.

$$U=\frac{V^2}{2g}=\frac{V^2}{64.34}. \qquad (1)$$

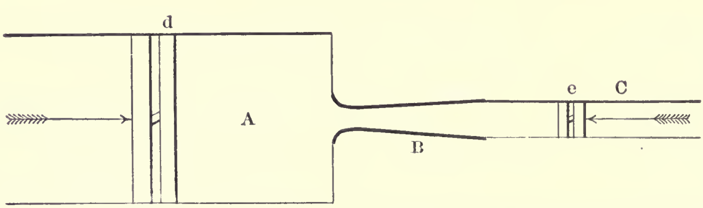


Fig. 92.—Ideal apparatus illustrating the flow of steam through a nozzle.

We have already seen that

$$U=J(H_1-H_2)=777.5\ \Delta H, \qquad (2)$$

whence

$$\frac{V^2}{64.34}=777.5\ \Delta H. \qquad (3)$$

Solving for V, the velocity of the steam leaving the nozzle,

$$V = \sqrt{64.34 \times 777.8\, \Delta H} = 223.6\sqrt{\Delta H}. \quad . \quad . \quad . \quad (4)$$

201. Form of the Turbine Nozzle. As the steam flows through the nozzle it increases in velocity and volume and diminishes in pressure. The area at any section is directly proportional to the specific volume of the steam and inversely proportional to its velocity. If the cross-sectional areas at a series of points in the nozzle be computed it will be found that the areas diminish at first until the pressure in the nozzle becomes about 58 per cent of the initial absolute pressure of the steam, and from that point onward the areas again begin to increase. The point of minimum section is known as the throat of the nozzle. The quantity of steam discharged by the nozzle obviously depends on the area of the throat or minimum section provided the steam is discharged into a region in which the pressure is less than 58 per cent of the initial steam pressure.

In order to have a nozzle discharge steam with a minimum of turbulence and friction, it is advisable that the acceleration of the body of steam contained within it shall be constant. Such a constant acceleration requires of course that the amount of heat energy transformed into work between any two cross-sections shall be proportional to the distance between these sections. A nozzle of the proper form may therefore be computed in the following manner:

First, having given the initial pressure and quality of the steam, its entropy should be found. Second, choose a series of pressures whose saturation temperatures differ by an approximately constant amount. Third, from the known entropy of steam during its adiabatic expansion, compute the total heat and specific volume of the steam at these several pressures. Fourth, compute the heat drop (i.e., the quantity ΔH) for each of the several pressures. Fifth, compute the resulting velocity at each of the several pressures. Sixth, from the velocity and specific volume of the steam at each of the several pressures, determine the proper cross-sectional area of the nozzle. Seventh, compute the diameter of the section for each of these several areas. Eighth, make the distance of each section from the inlet end of the nozzle proportional to the heat drop. These computations may be made from Marks and Davis's Steam Tables or may be approximately determined from the total heat entropy diagram. It is usually more convenient, however, to make them by means of Peabody's temperature-entropy table. The method of performing the computations may be seen from the following example.

Required to design a turbine nozzle to discharge 10,000 pounds of steam per hour, the initial pressure of the steam being 175 pounds absolute and the initial superheat 96°. The nozzle discharges into a pressure

of 20 pounds absolute. Assume that the entering velocity of the steam at the mouth of the nozzle is 100 feet per second. From Peabody's temperature-entropy table, the nearest pressure is 175.3 pounds and the nearest superheat is 95°.7. The entropy is 1.62. The work can be most easily performed by tabulating it in the manner shown in Table X.

TABLE X

P	H	ΔH	U	Vel.	Sp. V.	A_1	A	D	L
175.3	1251.2	....	0.2	100	3.018	0.03018	12.07	3.93	0.0
169.0	1247.6	3.6	3.8	435	3.102	0.00714	2.856	1.91	0.126
162.8	1244.0	7.2	7.4	608	3.190	0.00525	2.100	1.64	0.259
156.8	1241.4	10.8	11.0	741	3.280	0.00442	1.768	1.501	0.378
152.9	1238.0	13.2	13.4	818	3.343	0.00409	1.636	1.443	0.469
134.5	1226.3	24.9	25.1	1120	3.605	0.00329	1.316	1.296	0.872
117.9	1214.7	36.5	36.7	1354	4.079	0.00301	1.204	1.340	1.278
101.6	1201.8	49.4	49.6	1574	4.566	0.00290	1.160	1.217	1.730
89.6	1191.2	60.0	60.2	1735	5.018	0.00290	1.160	1.217	2.10
77.6	1179.4	71.8	72.0	1896	5.610	0.00296	1.184	1.230	2.52
67.0	1167.7	83.5	83.7	2042	6.384	0.00312	1.248	1.251	2.92
57.5	1155.9	95.3	95.5	2183	7.290	0.00334	1.336	1.304	3.34
49.19	1143.9	107.3	107.5	2316	8.368	0.00362	1.448	1.360	3.76
41.84	1131.6	120.6	120.8	2458	9.639	0.00392	1.568	1.415	4.22
35.32	1119.4	131.8	132.0	2567	11.16	0.00435	1.740	1.490	4.61
29.82	1106.9	144.3	144.5	2687	12.99	0.00482	1.928	1.570	5.05
24.97	1094.3	156.9	157.1	2800	15.18	0.00543	2.172	1.665	5.49
20.02	1079.0	172.2	172.4	2935	18.46	0.00629	2.516	1.792	6.02

In the first column, headed P, will be found the successive pressures for which the dimensions of the nozzle are to be computed. In the column headed H will be found the total heat of the steam at the given pressure, and entropy 1.62, as obtained from the temperature-entropy table. In the column headed ΔH will be found the difference between the initial total heat and the total heat at the pressure given. In the column headed U will be found the heat drop plus the initial kinetic energy of the steam in B.T.U. In the column headed Vel. will be found the velocity of the steam. In the column headed Sp.V. will be found the specific volume of the steam as taken from the steam tables. In the column headed A_1 will be found the area of a nozzle in square feet, per pound of steam flowing per second. In the column headed A will be found the actual area of the nozzle in square inches to pass 10,000 pounds of steam per hour. In the column headed D will be found the diameter of the nozzle in inches, and in the column headed L, the length in inches from the inlet end of the nozzle to the section having the diameter given. The following formulæ will be used in making the various computations:

$$\Delta H = H_1 - H_2 = 1251.2 - H_2,$$

$$U = \Delta H + \frac{V_1^2}{J\,2g} = \Delta H + \frac{100^2}{778 \times 64.36} = \Delta H + .2,$$

$$\text{Vel.} = 223.6\sqrt{U},$$

$$A_1 = \frac{\text{Sp.V.}}{\text{Vel.}},$$

$$A = A_1 \times 144 \times \frac{10000}{3600},$$

$$D = \sqrt{\frac{A}{0.7854}},$$

$$L = K\,\Delta H,$$

in which K is a constant so chosen as to make the nozzle of reasonable length. The form of the nozzle so computed is illustrated in Fig 93.

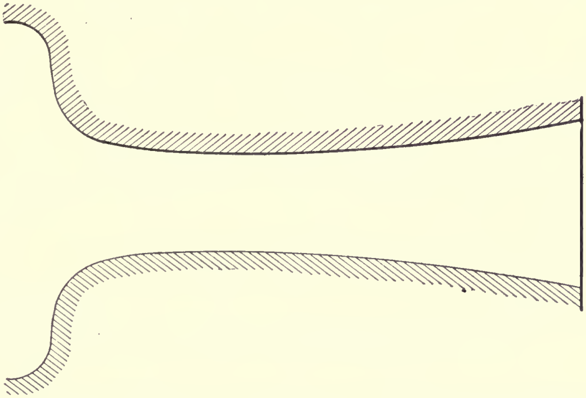


Fig. 93.—Form of turbine nozzle giving constant steam acceleration.

202. Alternate Methods of Designing a Turbine Nozzle. It will be seen that the work of computing the exact form of a nozzle which will give constant acceleration to the steam becomes laborious when a temperature-entropy table is not available. Consequently, steam turbine nozzles are usually designed by finding the area of the throat and of the mouth of the nozzle and making the nozzle of the form shown in Fig. 94. The radius of the entering portion should be made equal to the diameter of the throat in case a circular nozzle is employed. The divergent portion of the nozzle is a frustrum of a cone, the elements of which make an angle of about 5° with the axis. Sometimes the throat is made straight for a distance equal to one-half its diameter as shown

in Fig. 95. Either of these forms gives a nozzle of high efficiency, although it is not reasonable to suppose that the efficiency would be as high as in the case of a nozzle designed to give constant acceleration to the steam. The work of designing a nozzle like that in Fig. 94 may be seen from the

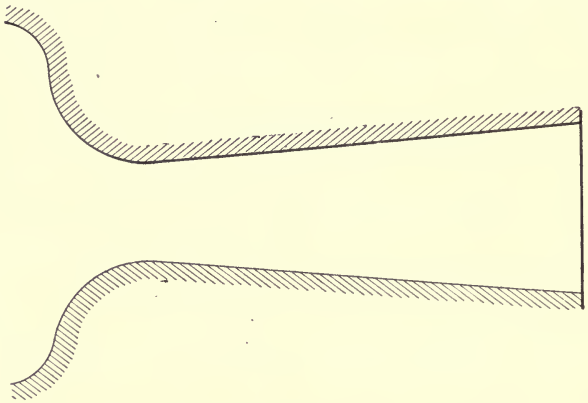


Fig. 94.—Turbine nozzle of the usual form.

following example, in which quantities from Mark's and Davis' Steam Tables are used:

Design a nozzle taking dry and saturated steam at 100 pounds and discharging 1 pound per second, against a pressure of 20 pounds absolute.

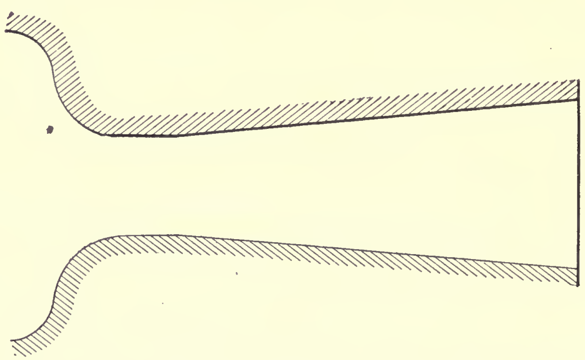


Fig. 95.—Another form of turbine nozzle, having a cylindrical throat.

The entropy of steam of 100 pounds pressure is 1.6020. Since the expansion in the nozzle is adiabatic, the entropy of the steam coming from the nozzle will be the same. The entropy of the liquid at 20 pounds absolute is 0.3355. The difference, or 1.2665, is the entropy of vaporization of the wet steam coming from the nozzle. The quality of the steam coming

from the nozzle may be found by dividing this quantity by the entropy of vaporization of dry steam of 20 pounds pressure and is

$$\frac{1.2665}{1.3965}=90.8 \text{ per cent.}$$

The total heat of steam of 100 pounds pressure is 1186.3 B.T.U. The total heat of the wet steam at 20 pounds pressure is found by the formula

$$H_2=h+qL \text{ and is } 196.1+90.8\times 960.0=1068.1 \text{ B.T.U.}$$

The difference, or 118.2 B.T.U. is the quantity of heat transformed into kinetic energy in the nozzle, a quantity previously designated by the symbol ΔH. Substituting in the formula

$$V=223.6\sqrt{\Delta H},$$

we will have 2435 feet per second for the velocity of the steam issuing from the nozzle. The specific volume of the steam may be found from its quality and will be $20.08\times 0.908=18.23$. Dividing this by the velocity of the steam, we will have for the area of the mouth of the nozzle required to discharge 1 pound of steam per second, 0.0075 square feet or 1.08 square inches.

The pressure of the steam in the throat of this nozzle will be 58 pounds. We may by the process already employed, find the area of the throat of the nozzle when it is required to discharge 1 pound of steam per second. The entropy of the liquid at 58 pounds is 0.4242. The entropy of vaporization will be 1.1178. The quality of steam will be

$$\frac{1.1178}{1.2218}=96.4 \text{ per cent.}$$

The total heat will be

$$259.8+916.5\times 96.4=1143.3.$$

The heat drop will be

$$1183.3-1143.3=40 \text{ B.T.U.}$$

The velocity of the jet at the throat will be

$$223.6\sqrt{\Delta H}=1413 \text{ ft. per sec.}$$

The specific volume of the steam passing the throat of the nozzle will be

$$96.4\times 7.45=7.13.$$

The area of the throat will be

$$\frac{7.13\times 144}{1413}=0.505 \text{ sq.in.}$$

203. Efficiency of Turbine Nozzles. The efficiency of a turbine nozzle is found by dividing the kinetic energy actually realized by the energy theoretically developed from the given pressure drop. On account of the friction of the steam against the walls of the nozzle, and also on account of the eddying produced when the nozzle is of improper form, the velocity of the issuing steam, the quantity of steam discharged and the efficiency of the nozzles are reduced. An improperly designed nozzle may give an efficiency as low as 90 per cent and a velocity and quantity of discharge of about 95 per cent of the theoretical value. A properly designed nozzle expanding steam between the pressure limits for which it was designed, ought to give an efficiency of more than 97 per cent, and such an efficiency has been realized by a nozzle of the form shown in Fig. 95.

204. The Design of Turbine Vanes. After the steam has been expanded in the nozzle, it is necessary to extract its kinetic energy by causing it

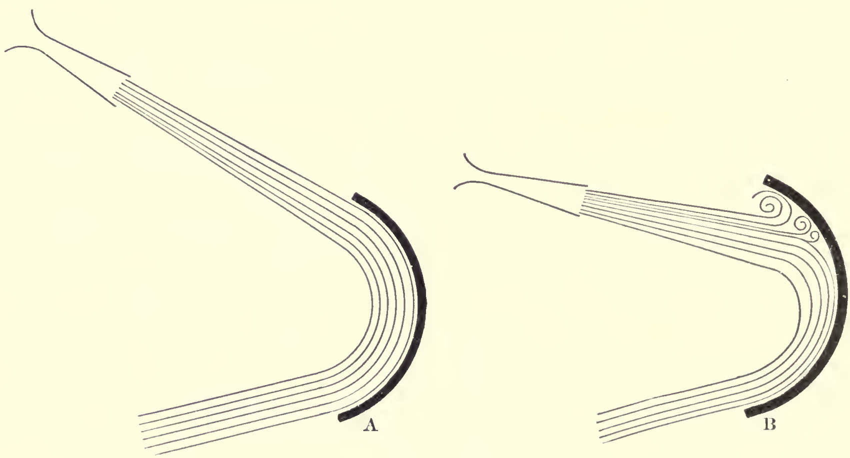


FIG. 96.—Properly and improperly formed vane surfaces.

to strike upon a moving surface. Its energy is usually utilized by causing it to strike upon a series of vanes (also termed blades and buckets) fixed upon the rim of a revolving disk or drum (which is often termed a rotor). The form and motion of the surface upon which the jet strikes should be such that the steam will be taken up smoothly and brought to some lower velocity without shock or eddying. This result will be achieved if the surface of the vane is suitably curved, and the steam enters the vane in such a manner that the direction of its motion relative to the vane is tangent to the curved surface of the vane at the entering edge, as shown in Fig. 96 *a*. If, however, the vane is not properly curved, or if the jet strikes obliquely upon its surface, there will be more or less

eddying of the steam at the point of impact and some of the kinetic energy developed in the nozzle will be lost by being retransformed into heat. The effect of such oblique impact may be seen in Fig. 96 *b*.

Since the vanes are moving as the steam enters them, the effect of this motion must be considered. Referring to Fig. 97, line *a–b* represents by its length and direction the absolute velocity and direction of motion of the entering jet of steam. Line *c–d* represents by its length and direction the velocity and direction of motion of the moving vanes. The velocity of the jet of steam relative to any point in the vanes will therefore be represented by the line *a–e*, which is the third side of the triangle whose second side, *b–e*, is parallel and equal to *c–d*. If the surface of the vane has the form shown by the curve *f–g*, it is apparent that the jet will strike the vane tangentially, that the steam will be taken up smoothly, without shock, and its direction of motion changed in the manner shown by the arrow. Except for the effect of friction, the relative velocity of the jet and the vane will remain unchanged, and the steam will leave the vane with a velocity relative to the vane which amount and direction are represented by the length and direction of the line *g–h* (which is equal in length to *a–e*). The absolute velocity of the steam leaving the vane will be represented by the line *g–i*, which is the third side of a triangle whose second side is *h–i* (*h–i* being equal and parallel to *c–d*). In consequence of the resulting change in the absolute velocity of the steam its kinetic energy will be reduced, the amount of energy imparted to the vanes in each second being equal to the difference between the initial and final kinetic energy of the steam discharged per second by the nozzle.

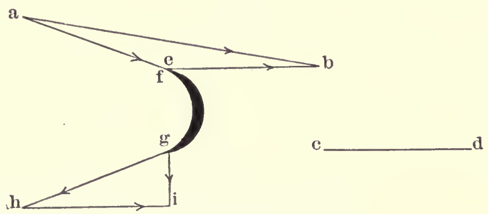


Fig. 97.—Velocity diagram.

205. Classification of Impulse Turbines. Impulse turbines are classified as **single stage** and **multi-stage**. A **single stage turbine** is one in which the entire pressure drop occurs in one set of nozzles. A **multi-stage turbine** is one in which only a portion of the pressure drop occurs in each set of nozzles, the steam flowing successively through two or more sets of nozzles. An impulse turbine may use one row of vanes per stage to take up the energy of the jet, or it may have two or more rows of moving vanes with stationary guide vanes interposed between each pair of moving rows. The De Laval turbine is a single stage turbine with a single row of moving vanes. The Kerr turbine is a multi-stage machine with a single row of buckets in each stage. The Curtis turbine is a multi-stage machine having several rows of vanes per stage. The simplest

form is, of course, the De Laval turbine, employing a single pressure stage and a single row of vanes. In order that the turbine shall have a high efficiency, it is necessary that the form and speed of the vanes shall be such that the steam is brought almost to complete rest. This requires that the vanes shall travel at very nearly one-half the velocity of the steam when only one row of moving vanes is employed. Since the velocity of the steam jet ranges from 2700 to 4500 feet per second, it will be seen that the vanes of the De Laval turbine must travel at peripheral speeds from 1300 to 2200 feet per second, in order to realize the full efficiency which it is possible to obtain from the steam. The De Laval turbine is so constructed that the vanes may run at these high speeds. The rotational speeds resulting are, however, entirely too high for driving

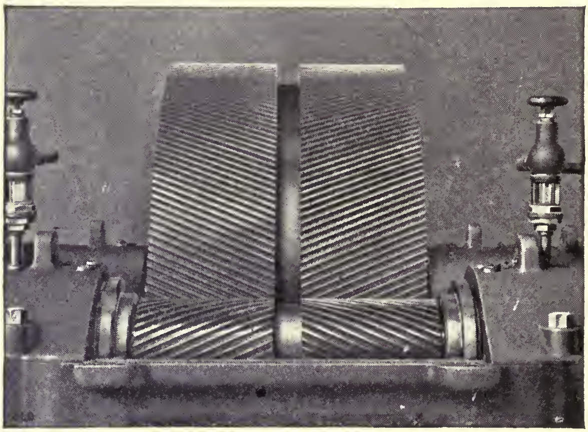


Fig. 98.

ordinary machinery, and accordingly, double helical gearing like that shown in Fig. 98 is employed to reduce the speed to a suitable value. Since it is difficult to construct such turbines in units of large size, the single stage De Laval turbines are built only in units of comparatively small power.

It is apparent, that if some method can be employed whereby the speed of the moving vanes may be reduced without reducing the efficiency of the turbine, it will be desirable to dispense with speed reduction gearing. This may be accomplished by reducing the velocity of the steam jet or by using several rows of vanes in such a manner that each row abstracts only a portion of the velocity of the steam jet. Both of these methods of speed reduction may be and often are combined in the same machine. When more than one row of moving vanes are employed, the steam leaving the first set of moving vanes is directed by a set of stationary vanes against a second set of moving vanes. The

steam may then be directed by a second set of stationary vanes against a third set of moving vanes, and this action repeated as many times as may be necessary. When this is done, each of the moving vanes extracts from the steam a portion of its velocity, and the peripheral velocity of the vanes may be greatly reduced. The use of several rows of vanes in this manner considerably decreases the efficiency of the turbine on account of the eddying resulting from the repeated reversals of the direction of the current of steam, and the friction resulting from the increase in the vane surface. On the other hand, the peripheral velocities obtained are so much smaller than those gotten when a single row of vanes is used, that it is practicable to construct this type in the largest sizes and to use it for driving many kinds of machinery without reduction gearing.

206. Design of Vanes for a Multi-stage Impulse Turbine. In designing the vanes for a turbine of this type, a velocity diagram may be used similar to that used for turbines employing a single row of moving vanes. Such a diagram is shown in Fig. 99, in which *a–b* represents the direction and velocity of the steam coming from the nozzle, *b–c* is equal to the velocity of the moving vanes, *a–c* is the relative velocity of the steam entering the first set of moving vanes, *c–d*, which is equal to *a–c*, is the relative velocity of the steam leaving the first row of moving vanes, *d–e* equals *b–c*, *c–e* is the absolute velocity of the steam leaving the first set of moving vanes, *e–f* represents the velocity and direction of the steam leaving the first set of stationary vanes, *f–g* equals *b–c*, *e–g* is the relative velocity of the steam entering the second row of moving vanes, *g–h* equals *e–g* and is the relative velocity of the steam leaving the second set of moving vanes, *h–i* equals *b–c*, and *g–i* represents the absolute velocity of the steam leaving the second set of moving vanes. The remainder of the diagram determines in like manner the velocity and direction of the current of steam as it passes through the third and fourth set of moving vanes.

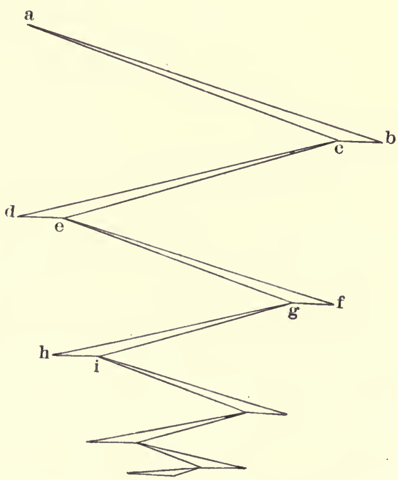


FIG. 99.—Velocity diagram for four rows of moving vanes.

The form of velocity diagram shown in Fig. 99 makes no allowances for friction. If it is assumed that the steam in its passage through a set of vanes loses a certain per cent of its velocity by friction, then the

velocity of the steam coming from the set of vanes will be less than the velocity of the steam entering that set by the per cent of loss due to friction. The diagram may therefore be modified to take account of friction in the following manner: Assume that 10 per cent of the velocity is lost in friction each time the current of steam passes through a set of vanes. Then the diagram will have the form shown in Fig. 100, in which *c–d* is 90 per cent of *a–c*, *e–f* is 90 per cent of *c–e*, *g–h* is 90 per cent of *e–g* and so on for the remainder of the diagram.

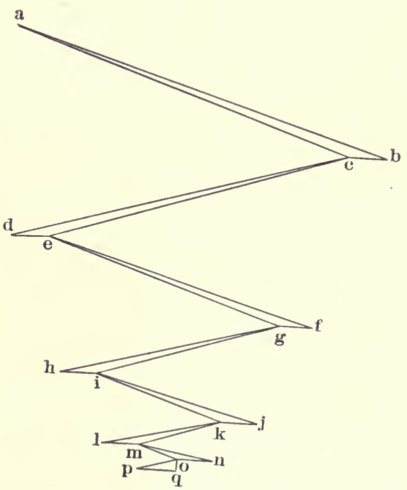


Fig. 100.—Velocity diagram corrected for friction loss.

The vanes are usually of the form shown in section in Fig. 101*a*. The surface *a–e–d* upon which the steam strikes, is usually the arc of the circle, although it may be any smooth curve, and the back *a–b–c–d*, consists of a circular arc *b–c* tangent to two straight lines *a–b* and *c–d*. Assume the steam to enter in such a manner that its real direction of motion is shown by the lines. The form of the blade is then made such that the line *a–b* is made parallel with the lines. It follows that the steam cannot enter the surface of the vane tangentially, but strikes it in the manner shown in Fig. 101 *a*. While this is somewhat objectionable, it would be more objectionable for it to strike upon the back of the vane in the manner shown in Fig. 101 *b*, and so retard the forward motion of the blade. The current of steam leaves the vane in the direction shown by the cover lines which are tangent to the face *a–e–d* at point *d*. In the case of a stationary vane, a similar form is to be employed, the steam entering in a direction tangent to the back of the vane and leaving it in a direction tangent to the face. The forms of the van are thus derived directly from the velocity diagram.

After the steam leaves the nozzles of an impulse turbine, its pressure remains constant. In consequence of its action upon the moving vanes, its velocity is reduced as it passes from row to row of blading, as shown by the arrow in Fig. 101 *c*. If there were no fluid friction, the specific volume of the steam would remain constant in passing through the blading of any stage. Since, however, the heat content of the steam is increased by friction and eddying, its specific volume will slightly increase as the steam passes the successive rows of vanes. It is neces-

sary that the area of the passage through which the current of steam flows shall always be sufficient to carry the entire quantity of steam flowing, at the velocity which it has while passing the point in question. In flowing through the first set of vanes, the steam has a very high velocity. In passing through the last set of vanes, it has a comparatively low velocity and its specific volume is slightly larger. Consequently, the area of the steam passage through the last set of vanes must be much greater than that through the first set. This is usually accomplished by making each successive row of vanes longer than the pre-

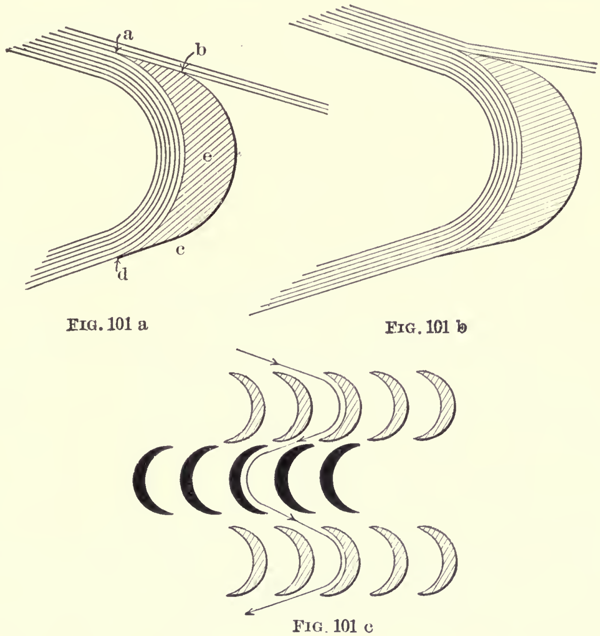


FIG. 101.

ceding, the heights of vanes being practically inversely proportional to the velocity of the steam passing through them.

207. Determination of the Pressure and Energy Drop in Multi-Stage Impulse Turbines. When steam passes through a multi-stage turbine its entropy does not remain constant. In passing through the nozzles it undergoes adiabatic expansion, but when it issues into the vanes, eddying and friction transform a portion of its kinetic energy into heat, increasing its entropy. Let ΔH equal the heat drop resulting from adiabatic expansion between the initial and condenser pressure. Let E equal the efficiency of the turbine, expressed as a per cent of the efficiency of a perfect turbine, a quantity which may be found from the expression

$$E = \frac{2547}{S \times \Delta H},$$

in which S is the probable steam consumption of the turbine in pounds per brake horse-power per hour. Let n equal the number of stages. Then, if E be very nearly unity, and the steam passes through the turbine without appreciable increase in entropy, the heat drop per stage will be $\frac{\Delta H}{n}$. The pressure of the steam entering any stage may be found by subtracting the quantity $\frac{\Delta H}{n}$ from the total heat of the steam, as it enters the nozzles of the preceding stage, and finding, by means of a temperature-entropy table, or a total heat-entropy diagram, the pressure of steam having the total heat so found, and the original entropy. For instance, if the heat content of steam entering any stage is 1161 B.T.U., its entropy is 1.68 and the quantity $\frac{\Delta H}{n}$ is 50 B.T.U., then the heat content after expansion is 1111 B.T.U. The pressure of steam whose total heat is 1111 B.T.U. and whose entropy is 1.68, is found from the temperature-entropy table to be 17.52 pounds per square inch, which is the pressure of the steam entering the next stage.

Usually the efficiency of a turbine will range from 55 to 80 per cent. In such a case the heat drop per stage may be found by the empirical equation

$$h=\frac{\Delta H}{2}\left(1+.00057\left(\frac{n-1}{n}\right)\Delta H(1-E)\right).$$

The heat content of the steam after expansion in the nozzles will be

$$H_1-h=H_2',$$

in which H_1 is the total heat of the steam entering the nozzles, and H_2' is the total heat as it leaves the nozzles. The pressure of the steam leaving the nozzles is, of course, fixed by the quantity H_2' and the entropy of the steam as it enters the nozzles.

The quantity of heat restored to the steam by eddying and friction is equal to

$$h-\frac{E\Delta H}{n}$$

and the heat content of the steam after it passes through the blading will be

$$H_1-\frac{E\,\Delta H}{n}=H_2.$$

The entropy of the steam entering the next stage will be fixed by the pressure of the steam already found, and the quantity H_2.

The following problem will illustrate the method of designing a multi-stage turbine: Assume that the initial steam pressure is 164.8 pounds per square inch, that the final pressure is 1 pound per square inch, that the steam is initially dry and saturated, that the number of stages is 2 and the efficiency E is 60 per cent. The initial entropy is 1.56. The total heat at the same entropy and terminal pressure is 87.1 B.T.U. The difference, or 322.2 B.T.U., is the heat drop, ΔH. The heat drop per stage will then be

$$\frac{322.2}{2}\left(1+.00057\left(\frac{2-1}{2}\right)322.2(1-.60)\right)=167.0 \text{ B.T.U.}$$

The heat content of the steam as it issues from the nozzle of the first stage will therefore be $1193.3-167.0=1026.3$ B.T.U. Steam of 1.56 entropy having this heat content has a pressure of 16.86 pounds per square inch. The nozzles of the first stage are therefore designed to discharge the required quantity of steam and to operate

between 165 pounds and 16.86 pounds per square inch. After passing through the blading the heat content of the steam will be

$$1193.3-\frac{.60\times 322.2}{2}=1096.6 \text{ B.T.U.}$$

The entropy of steam of 16.86 pounds pressure and having a total heat of 1096.6 B.T.U. is 1.663. The total heat of the steam issuing from the nozzles of the second stage will be $1096.6-167.0=929.6$ B.T.U. The pressure of steam of this heat content and 1.633 entropy is 1 pound. The nozzles of the second stage will be designed to pass the same quantity of steam as will be passed by the nozzles of the first stage and to operate between the pressure limits of 16.36 pounds and 1 pound absolute. Since the steam enters the nozzles of the second stage with low velocity and its final velocity as it issues from these nozzles is the same as it was when it issued from the nozzles of the first stage, and since the specific volume of the steam at low pressure is much greater than it was at the high pressures used in the first stage, the cross-sectional area of the nozzles of the second stage will be much larger than that of those of the first stage.

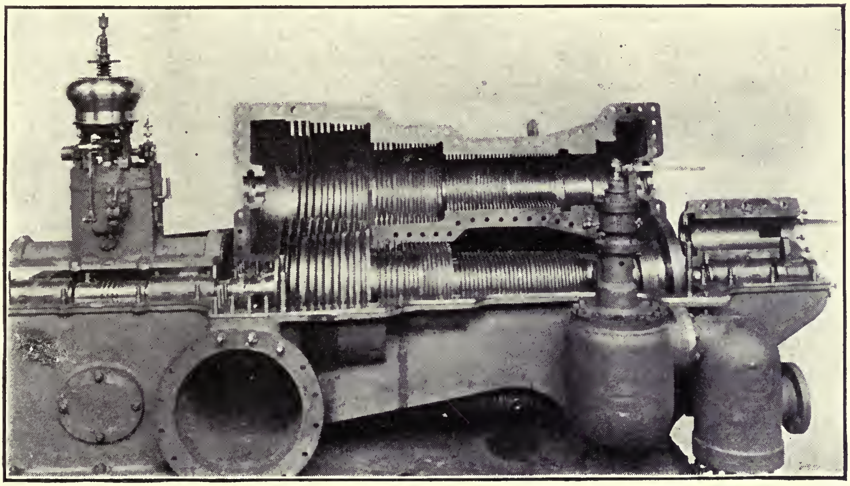


FIG. 102.—Westinghouse-Parsons turbine of an early type, with the top of the casing removed.

208. The Impulse Reaction Turbine. When a reaction turbine of the Avery type is used, the centrifugal force of the steam contained in the revolving element increases its pressure and consequently its velocity, as it issues from the nozzle. This increased velocity in turn increases the centrifugal force, which, in its turn, again increases the velocity of the issuing steam. In order that such a turbine shall run with perfect efficiency, transforming all of the available heat into work, it is therefore necessary that the arms shall revolve at infinite speed, which is of course impossible. No arrangement of parts or division into stages will eliminate the necessity of operating the rotor at excessive speeds in order

to secure reasonable efficiency. Consequently, pure reaction turbines are no longer built. By combining the principles of the impulse and the reaction turbines, as is done in the Parsons turbine, illustrated in Fig. 102, a very efficient mechanism may be obtained. The blading of such turbines consists of alternate rows of stationary and moving vanes, as shown in Fig. 103. The moving vanes are cross-hatched and the stationary vanes are blackened in the figure. On account of the difference in pressure at the two ends of the turbine, the steam flows through the blading with high velocity, expanding as it flows. The passages between the stationary blades act as fixed nozzles, while those between the moving vanes act as moving nozzles. The steam gains in absolute velocity in passing through a row of stationary vanes, and impinges upon the following row of moving vanes, driving them forward by its impulse, As it passes through the row of moving vanes, it gains in relative velocity. and again drives them forward by the reaction of its discharge. The velocity diagram of such a turbine is shown in Fig. 104. The velocity of the steam as it issues from a row of fixed vanes is represented by the line a–b, b–c represents the velocity of the moving vanes, and a–c represents the relative velocity of the steam and the vanes. The relative velocity of the steam issuing from the moving vanes is c–d, and its absolute velocity is represented by the line c–e, d–e being equal to b–a. In consequence of the pressure drop, this velocity increases to the value fg in passing through the next row of fixed vanes. The steam continues to pass from row to row of vanes, increasing in absolute velocity while passing through the stationary vanes, and decreasing in absolute, but increasing in relative velocity, while passing through the moving vanes.

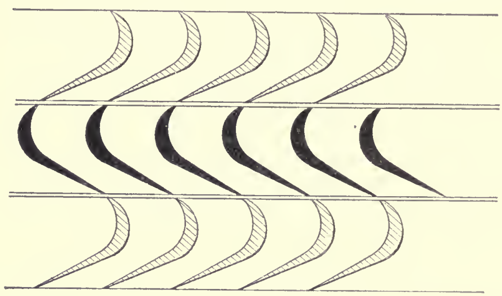


FIG. 103.—Vane sections for an impulse-reaction turbine.

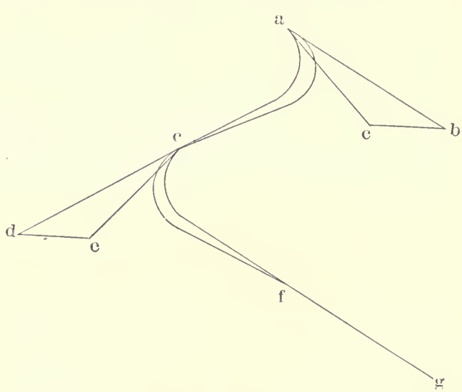


FIG. 104.—Velocity diagram for an impulse-reaction turbine.

The design of an impulse reaction turbine is a much more complicated and tedious process than the design of an impulse turbine. For the proper methods of designing such a turbine, the reader is referred to any of the standard works on Steam Turbines. An excellent treatment of the subject will be found in Roe's "Steam Turbines," pages 54 to 72.

Many special forms of the steam turbine are in use which cannot be described in a work of this kind. In principle they all belong to one or another of the classes described, but differ in their mechanical arrangements.

209. Methods of Governing the Steam Turbine. Three methods are in use for governing the steam turbine. The first method consists in throttling the pressure of the steam at entrance. This reduces the quantity of steam which flows through the nozzles, and in a greater degree it reduces the work performed by the turbine. Its effect is somewhat similar to the effect of a throttling governor upon the steam engine. Reducing the pressure of the entering steam changes the pressure in each of the stages of the turbine, changes the velocity with which steam enters the buckets, and reduces the efficiency of the turbine, for if the turbine is properly designed for full steam pressure, the form of the nozzle, the shape of the blading, and the velocity of the rotating parts will all be incorrect for the reduced pressure which results from throttling. The turbine will therefore operate at much lower efficiency at low load than it does at full load.

The second method of governing the steam turbine is to admit the steam intermittently. A balanced valve in the pipe supplying steam to the turbine is opened an instant and then quickly closed, the action occurring at regular intervals. While it is open steam passes freely through the turbine, and its operation is normal. When it is closed, the pressure of the steam throughout the turbine immediately drops to back pressure and the turbine continues to revolve until the next blast of steam is admitted. While the turbine is receiving steam, it is working at maximum efficiency. While it is not receiving steam, it is revolving without loss except that due to the passage of the blades through the steam at very low pressure. The blast of steam is admitted to the turbine from 60 to 150 times per minute, the duration of the blast being determined by the governor. When the load is light, the pressure variation in the first stage will have the form shown in Fig. 105 *a*; when the load is heavy, it will have the form shown in Fig. 105 *b*; and when the turbine is overloaded, the steam is admitted continuously. This method of regulation is much superior to regulation by throttling, and the great inertia of the revolving parts prevents any perceptible variation in speed, in spite of the intermittent character of the force applied to the turbine blading.

A third method of turbine governing consists in providing each stage with the same number of nozzles, or groups of nozzles, and admitting full steam to as many groups as may be necessary in order to carry the

load. Turbines of large sizes are ordinarily equipped with six or eight groups of nozzles per stage. At low loads only a small part of the nozzles will be open to the admission of steam, while at heavy loads almost all the nozzles will be open, and when the turbine is running at the limit of its capacity all of the nozzles will be in operation. This method of governing has no advantage over the system of intermittent admission, and cannot be applied to impulse reaction turbines. It is, however, easy to apply it to pure impulse turbines, particularly when the number of stages is small. Most impulse turbines are governed in this manner, while impulse reaction turbines are usually governed by intermittent admission.

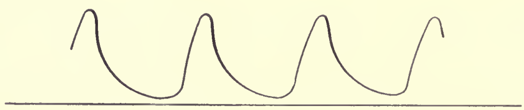


Fig. 105 *a*.

Fig. 105 *b*.

210. Efficiency of the Steam Turbine. The steam turbine operates upon the Rankine cycle with complete expansion of the steam. Since it uses a much more efficient cycle than the steam engine, it might be inferred that the steam turbine would be much more efficient than the steam engine. Such is not the case. The efficiency of the two types of apparatus, as measured by the heat consumption required per brake horse-power per hour, is practically the same, the steam engine being slightly more economical than the steam turbine for the same temperature range. Since the form of cycle employed in the steam engine does not utilize the lower part of the temperature range to good advantage, it follows that the steam engine can use high pressure steam more efficiently than the steam turbine, while on the other hand the steam turbine can use low pressure steam more efficiently than the steam engine. Consequently, when the steam is supplied to a steam engine at high pressure and superheat, and this engine discharges this steam at or about atmospheric pressure into a steam turbine which expands it down to the lowest practicable condenser pressure, the efficiency of the entire plant will be greater than it would be were a steam engine or a steam turbine alone employed. Such a combination of units has been adopted in the Interborough Power Station in New York city, in which a 7500 K.W. compound engine discharges its exhaust into a steam turbine of approximately equal power. The efficiency of the combined unit is about 25 per cent greater than the efficiency of the engine when it is operated as a condensing engine.

The following table will serve to show the efficiency of large steam engines and steam turbines for different conditions of operation:

Maker.	Location of Plant.	Load in K W.	Steam Pressure, Absolute	Superheat.	Back Pressure, Absolute.	Pounds Steam per K W Hour.	Pounds Steam per I.H.P.Hour.*	B.T.U. per K.W. Hour.	B.T.U. per I.H.P Hour.	Efficiency Ratio.†	Total Efficiency.‡
TURBINES											
Parsons	Dunstan, Eng.	6,257	204	176	0.44	11.95	8.02	14,960	10,400	76.4	24.5
Erste Brunner	....	6,000	191	194	0.88	12.58	8.44	15,570	10,450	79.2	24.4
Allgemeine Electricitäts Gasellschaft	Berlin (Moabit)	3,169	185	215	0.45	12.70	8.52	16,100	10,800	71.8	23.6
Bergman	....	1,545	195	201	0.68	12.97	8.71	16,200	10,890	73.6	23.4
Zoelly	Charlottenburg	2,052	200	202	0.75	13.05	8.76	16,300	10,940	73.9	23.2
Erste Brunner	Prague	2,000	161	118	1.03	13.84	9.29	16,470	11,050	79.6	23.1
General Electric Co. (Curtis Turbine)	Boston Edison Co.	5,195	189	142	0.58	13.52	9.07	16,560	11,110	72.2	22.9
Allis-Chalmers Co.	Richmond, Va.	4,328	186	108	0.96	14.02	9.41	16,700	11,210	76.0	22.7
Brown-Boveri	....	3,500	162	133	0.55	13.72	9.21	16,710	11,210	72.9	22.7
Westinghouse Co.	Brooklyn Rapid Transit Co.	11,601	192	114	1.03	14.23	9.55	16,970	11,390	75.1	22.4
DeLaval	Varberg	1,570	172	170	0.70	16.47	11.05	20,270	13,600	61.2	18.7
ENGINES											
....	Berlin (Moabit)	1,920	203	223	0.94	13.35	8.96	16,700	11,210	73.5	22.7
Westinghouse	New York Edison Co.	3,872	200	0	1.35	16.78	11.26	18,650	12,520	70.9	20.3
McIntosh and Seymour	Boston Edison	1,500	171	92	2.22	16.47	11.05	18,960	12,720	75.6	20.0
Allis-Chalmers	New York	5,496	190	0	2.41	17.82	11.96	19,520	13,100	72.9	19.4
COMBINATION											
Allis Engine / Curtis Turbine	New York	11,240	194	0	0.58	13.19	8.74	14,950	10,300	81.5	24.7

* On the assumption that the combined efficiency of the engine or turbine and generator is 90 per cent.

† This is the ratio of the actual efficiency as determined from the heat rate per indicated horse-power to the efficiency of the Rankine cycle for the same temperature range.

‡ The per cent of the total heat supplied actually transformed into indicated work.

PROBLEMS

1. Dry and saturated steam enters a turbine nozzle at a pressure of 100 lbs. per square inch. The nozzle discharges into a pressure of 2 lbs. per square inch. Find the theoretical velocity of the steam discharged. Ans. 3560 ft. per sec.

2. Dry and saturated steam at 150 lbs. enters a turbine nozzle. Find the pressure and velocity of the steam in the throat of the nozzle.

Ans. 87 lbs. and 1470 ft. per sec.

3. Design a turbine nozzle taking dry and saturated steam at 20 lbs. pressure and discharging it at 2 lbs. pressure. The entering velocity of the steam is 100 ft. per second. Use Peabody's temperature-entropy table if available and design a nozzle having constant acceleration.

4. Design a nozzle for the conditions in the preceding problem, making it of the form shown in Fig. 94, using the methods outlined in Art. 202.

5. A four-stage turbine takes steam at 150 lbs. pressure and 110° superheat. The back pressure is 1 lb. absolute. Find the proper pressure in each of the four stages, assuming that the efficiency of the turbine is 75 per cent.

Ans. 150 lbs., 54.5 lbs., 16.9 lbs., and 4.5 lbs.

CHAPTER XIII

CONDENSING MACHINERY

211. Classification of Condensers. There are in use with the steam engine two classes of condensers. In the first class of condensers the steam is brought into contact with a metallic surface, which is continually cooled by the application of cold water to the opposite side of the metal, and the condensed steam and the air which is mingled with the vapor in the condenser are removed by some form of pump, termed the **air-pump.** In the second class of condensers the condensing water and the steam are brought into direct contact, and it is necessary for the air-pump to remove not only the condensed steam and the air which it has brought over, but also the water of condensation and its entrained air. Condensers of the first class are known as **surface condensers.** Condensers of the second class are divided into **jet condensers, barometric condensers,** and **ejector condensers.**

212. Surface Condensers. In the surface condenser the cooling water is usually caused to flow through thin-walled metal tubes by means of a pump termed the **circulating pump.** These tubes may be made of any suitable kind of metal, such as copper, iron or brass, but tinned brass tubes are most usual. These tubes are placed within a metal shell, usually made of cast iron, into which the steam enters from the exhaust pipe of the engine. Every particle of steam coming in contact with one of these tubes is immediately condensed, and were no air present, the pressure of the steam would be that corresponding to the temperature of the outside of the condenser tubes, since as soon as each bit of steam is condensed by contact with a cold tube, it will leave about the tube a vacuum into which other steam will rush, so causing the condensation to be continuous. Since, however, the steam contains some air, as was explained in Art. 167, these tubes are surrounded by a rarefied atmosphere, through which the steam makes its way with some difficulty, so that the temperature of the steam in the condenser is higher than that of the surface of the condenser tubes.

One of the fundamental points of surface condenser design is to so arrange the condensing surfaces that the blast of steam which sweeps over them from the exhaust pipe will clear away the air surrounding them, and by the continual stirring up of the vaporous contents of the

condenser, bring every particle of steam as quickly as possible into contact with the condensing surfaces.

Since the steam which is condensed is continually bringing into the condenser quantities of air, it follows that if this air is not removed, the pressure within the condenser will continually increase and the efficiency and power of the engine will be correspondingly reduced, until finally no advantage will be obtained from the condenser, since the pressure in the condenser will be as great as the pressure of the atmosphere. In order to avoid this difficulty, the air must be removed from the condenser as fast as it is introduced. The pump which removes the air is called the air-pump. In case it removes air alone, and the water of condensation is removed by a separate pump, it is called a dry-vacuum pump. A condenser should be so arranged that as the steam flows through it, the air should be removed from that point of the condenser most distant from the entrance, since this will result in the removal of the maximum quantity of air in a given quantity of vapor. It will be understood that the air-pump not only removes air, but also the vapor or steam present in the condenser, and it is therefore desirable to have the air-pump draw its charge from that portion of the condenser which contains the largest proportion of air in the vapor.

213. Arrangement of Cooling Surface and Air-Pump. The tubes of a condenser are usually about $1/2$ to 1 inch in diameter, and the water flows through them from one end to the other. Condensers are commonly made in the manner shown in Fig. 106, which is a diagrammatic cross-section of a condenser. The steam enters from the exhaust pipe at A and as it enters, it encounters the cool tubes, where it is condensed. The current of steam passes to the left along the tubes and then back on the other side of the plate B, where the steam and condensed water is drawn by the air pump through the outlet C. The water enters the chamber D from the circulating pump and flows to the left through the tubes until it reaches the chamber E, from which it returns through the upper rows of tubes into the chamber F separated by the partition G from the chamber D and leaves by the outlet shown. It will be noted that the current of cooling water is opposite in direction to the current of steam. The effect of this is to increase the efficiency of the condenser, as will appear from the following.

The steam which enters the condenser at A carries with it some air. This comes in part by leakage through joints in the exhaust pipes and passages, in part by leakage around the piston rod and valve stems of the low pressure cylinder, and in part from air dissolved or intrained in the feed water. Since the condenser is open from end to end, there is only a very slight difference in pressure between the inlet and the outlet. The amount of air present in each unit of volume is much

greater, however, near the outlet than it is near the inlet, since the condensation of the steam occurs at all points through the condenser and is particularly rapid near the inlet. Since the pressure of the air is greater near the outlet on account of the greater amount of air present, the pressure of the steam and therefore its temperature, must be less near the outlet. In order to extract the heat most effectually from such a mixture of steam and air it is necessary to bring the coldest vapor into contact with the coldest condensing water and the hottest condensing water into contact with the hottest vapor. Hence, the most efficient surface condenser will be that in which the steam is admitted to the condenser at the point where the condensing water is discharged and the condensing water introduced at the point where the air pump takes its suction.

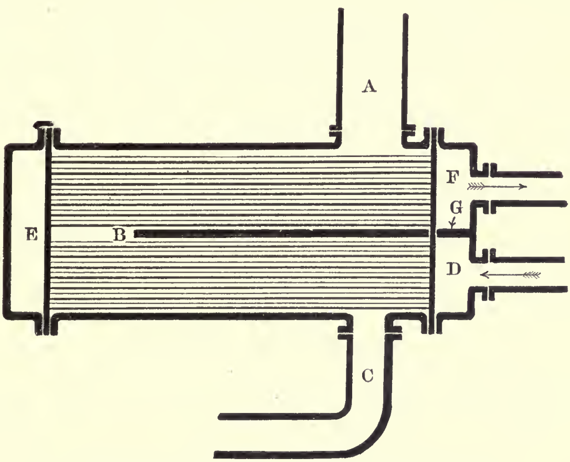


Fig. 106.—Section of a surface condenser.

214. Theory of the Surface Condenser. The following theory of the surface condenser is based on the assumption, which is not quite fulfilled in practice, that the air pressure in a condenser is infinitesimal, and the steam in all parts of the condenser is at the same temperature. When water passes through a tube surrounded by steam of a given temperature, the steam condenses upon the outside of the tube (provided the water is colder than the steam) and the water receives heat by the process, consequently increasing in temperature. Assuming a tube having a diameter c in feet, a length L, in feet, and through which water is flowing with the velocity V, in feet per second, we will have the water warmed at a rate depending upon the rate of heat absorption. It is known that the amount of heat transferred from steam to water under these conditions depends upon the difference in temperature between the steam and water. Let this difference in temperature be T, a variable, let T_s

be the temperature of the steam and T_w be the initial temperature of the water. Then the number of B.T.U. transferred from the steam to the water through each square foot of tube surface per second is equal to $K\ T$, where K is a constant to be determined experimentally.

Let us assume that we have within the tube a small volume of water whose length is dL and whose area is that of the cross-section of the tube. The weight of this small quantity of water will be

$$0.7854\ c^2\ dL \times 62.5 = 49.1\ c^2\ dL \text{ lbs.} \quad \ldots \ldots \quad (1)$$

If its temperature in a given short increment of time be increased by dT, and its specific heat is assumed to be unity, the amount of heat absorbed will be

$$49.1\ dT\ dL\ c^2 = \text{the heat absorbed.} \quad \ldots \ldots \quad (2)$$

The heat absorbed through the tube in the given increment of time, dt, will be equal to

$$3.1416\ c \times dL\ K\ T\ dt = \text{the heat transferred.} \quad \ldots \quad (3)$$

The heat transferred will, of course, equal the heat absorbed by the water, and we may so write them, or

$$3.1416\ c\ dL\ K\ T\ dt = 49.1\ dT\ dL\ c^2. \quad \ldots \ldots \quad (4)$$

Clearing, we have

$$\frac{dT}{T} = 0.064\ \frac{K}{c}\ dt. \quad \ldots \ldots \quad (5)$$

Integrating these expressions we have

$$\log_e T = 0.064\ \frac{K\,t}{c} + C. \quad \ldots \ldots \quad (6)$$

In order to find C, we may put t equal to zero, in which case the difference in temperature between the steam and the water will be $(T_s - T_w)$, since the water has received no addition of heat. From this we deduce that

$$C = \log_e (T_s - T_w), \quad \ldots \ldots \quad (7)$$

and

$$\log_e T = \log_e (T_s - T_w) - 0.064\ \frac{K\,t}{c}. \quad \ldots \ldots \quad (8)$$

It may be noted that since the difference in temperature T is continually diminishing as the time, t, increases, the expression dT is essentially negative, which throws the equation into the form given when properly written.

In order to find the temperature to which the water will be raised in a condenser or feed-water heater, we may write $\frac{L}{V}$ for t, which gives the length of time required for the water to traverse the given length of the tube. Substituting this value for t we obtain the temperature difference between the water and the steam after the water has traversed the condenser tube, and from this the rise in temperature and the quantity of heat absorbed by the water in traversing the tube.

215. Rate of Heat Transmission in Surface Condensers. Experiments by various engineers, according to Kent, give for the rate of transmission of heat through clean metal surfaces, from 0.09 to 0.18 B.T.U. per square foot per second. In the case of ordinary metal surfaces fouled by cylinder and saline deposits, the conductivity is about $1/3$ that for clean metal surfaces. If we substitute the value given above for K we will find that the equation reduces to the form

$$\log_e T = \log_e (T_s - T_w) - 0.004 \frac{L}{cV}. \quad . \quad . \quad . \quad . \quad (1)$$

This equation, for the purposes of computation, may be reduced to the form

$$\log T = \log (T_s - T_w) - 0.02 \frac{L}{dV}, \quad . \quad . \quad . \quad . \quad . \quad (2)$$

or to the form

$$\log T = \log (T_s - T_w) - 0.02 \frac{t}{d}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

in which T is the difference in temperature between the water and steam at any instant, t is the length of time during which the water has been flowing through the condenser in seconds, d is the diameter of the condenser tubes in **inches,** L is the length of the condenser tube in feet. V is the velocity in feet per second for the water flowing in the tubes, T_s is the temperature of the steam and T_w is the initial temperature of the water. The constant given above is that which is proper for brass condenser tubes under average condition. In the case of iron tubes the constant would be slightly less, and in the case of copper tubes slightly greater than 0.02.

If we plot from this equation the relation between the temperature of the water flowing in the condenser tubes and the length of time during which it has been flowing through the condenser, as is done in Fig. 107, we will find that the water starts at the temperature T_w and rapidly increases in temperature at first. As t increases, however, this rate of temperature increase becomes less and less, and the temperature finally approaches, but never reaches, T_s. Hence, no matter how much we may increase the area of the condensing surface, we cannot bring the temperature of the condensing water to the temperature of the steam. From this same figure it will also be seen that if the amount of water flowing through a condenser be diminished, the final temperature of

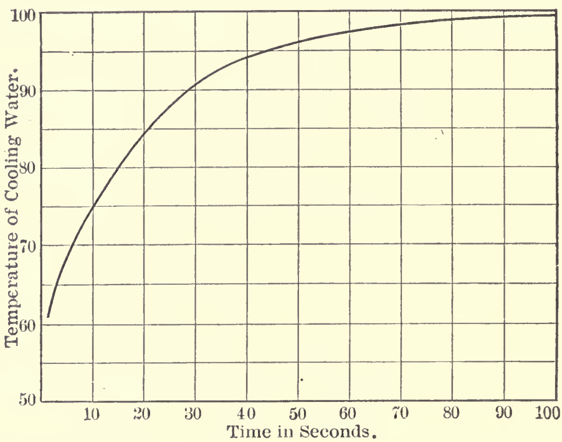


FIG. 107.—Relation between the temperature of the cooling water and the time it occupies in passing through the condenser.

the water will be increased. However, since this increase in temperature is not proportional to the decrease in the quantity of water flowing, the capacity of the condenser will be diminished. Fig. 108 shows the relation between the capacity of the condenser, in pounds of steam per square foot of cooling surface per hour, and the condensing water supplied per square foot of cooling surface per hour, at temperatures 60° and 30° below the steam temperature. An inspection of these curves shows that by increasing the quantity of water flowing, we can increase the quantity of steam which may be condensed without changing the size of the condenser, but doubling the quantity of circulating water will not double the quantity of steam condensed, although it will very largely increase it.

216. The Jet Condenser. In the case of the jet condenser which is illustrated in Fig. 109, steam is introduced into a receiver (which is

usually pear shaped in form) at the top. Into this receiver there is sprayed, usually by the suction created by the air-pump, a supply of water. This water being introduced in the form of fine spray exposes a large surface upon which the steam in the condenser quickly condenses. The mingled water of condensation and condensed steam are then withdrawn, together with the air which has been brought in by the steam,

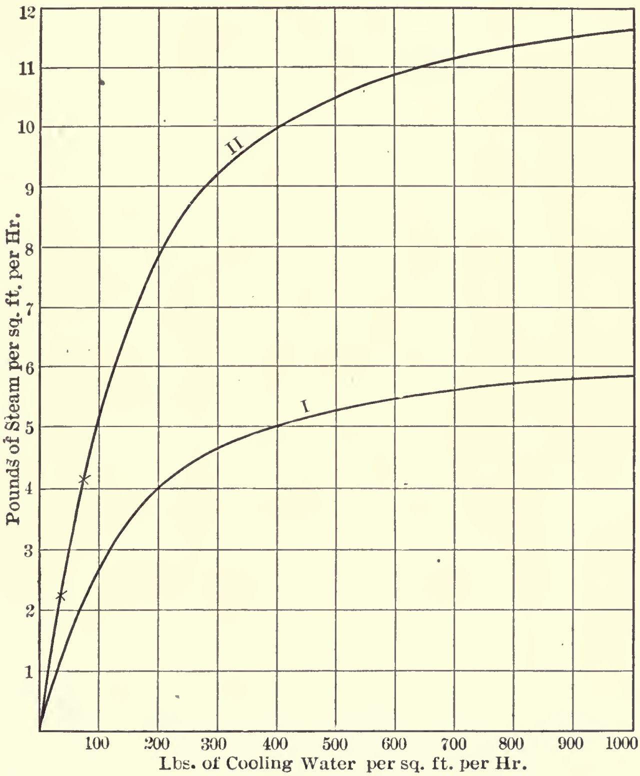


Fig. 108.—Relation between the water circulated and the capacity of the condenser.

Curve 1. For an initial temperature difference of 30° between the steam and cooling water.

Curve II. For an initial temperature difference of 60° between the steam and the cooling water.

and that given up by the condensing water under the combined influence of heat and vacuum. In the case of the jet condenser, the air-pump must be very much larger than in the case of the surface condenser, in order to attain the same vacuum. No circulating pump is required ordinarily with a jet condenser, the air-pump performing that service. The amount of air to be drawn away in a given time, in the case of a jet condenser

is several times that which must be drawn away in the case of a surface condenser of the same capacity, since a considerable proportion of the air in the jet condenser is that which is brought in by the condensing water. Where a high vacuum is to be maintained other forms of condensing apparatus are usually preferable to the jet condenser. This is also true in those cases where it is desirable to use the condensed steam as boiler feed, as for instance in marine work, and stationary power plant work when steam turbines are used. In the latter case, since the steam turbines do not require internal lubrication as do steam engines, the exhaust steam carries no oil and the condensate is suitable for boiler feed. In the case of the ordinary reciprocating engine, however, the cylinder oil in the exhaust steam is difficult to extract, and unless it is extracted, the water of condensation is not suitable for boiler feed.

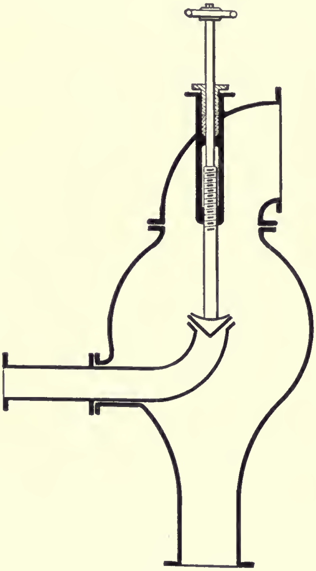


FIG. 109.—Section of jet condenser.

217. The Ejector Condenser. The ejector condenser is a type of condensing apparatus which depends upon the velocity of the stream of condensing water to carry away the air in the condenser. Such an apparatus is illustrated in Fig. 110. The steam enters the condenser through the pipe A. The condensing water is forced into the condenser under considerable pressure through the pipe B, and flows into the body of the apparatus at high velocity in the form of a hollow cone, the steam condensing upon its surface. The air associated with the steam is swallowed up in this stream of water and carried past the throat of the condenser in the form of innumerable bubbles. As the stream of condensing water and condensed steam descends through the tail pipe F, these bubbles are carried through, since the velocity of the water in the tail pipe is higher than the velocity at which the bubbles ascend. The condensing water, the condensed steam and the entrained air are finally discharged into a well at the bottom of the tail pipe. Many arrangements are in use for exposing a greater area of the condensing water to the action of the steam and for regulating the quantity of water used when the condenser is not operated at its capacity. It is not necessary that the condenser should be set up at an elevation in the manner shown, since the placing of a pump, preferably a centrifugal pump, in the tail pipe, will permit this type of condenser to be used when head room is limited.

218. The Barometric Condenser. The barometric condenser shown in Fig. 111 differs from the ejector condenser in that it does not depend upon the velocity of the condensing water to eject the air, and from the jet condenser in that the condensing water is not removed by the air-pump. The air is removed by a separate pump known as a dry vacuum pump, through the air pipe A. A **tail pipe** F, shown in the drawing, is

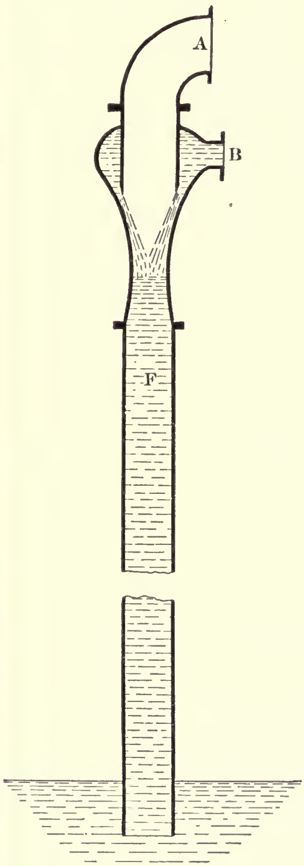


Fig. 110.—Diagram of an ejector condenser.

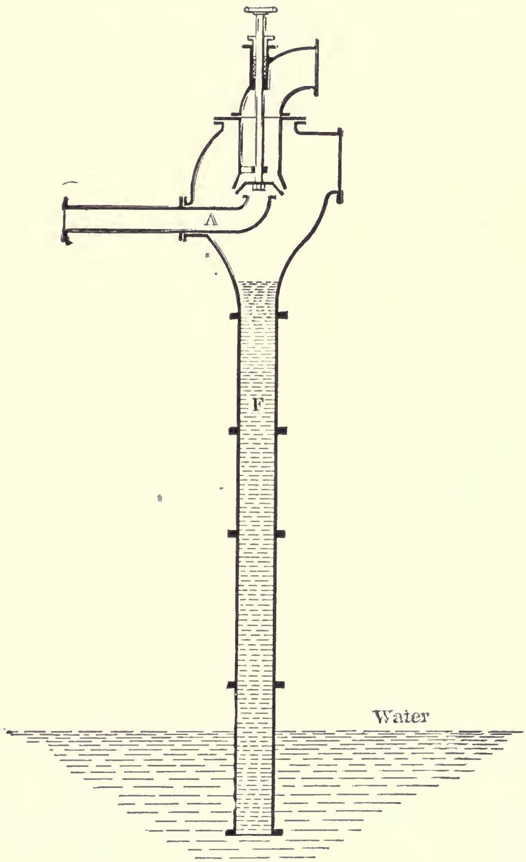


Fig. 111.—Section of a barometric condenser.

of use only to carry away the condensing water. The water rises to such a height in the tail pipe that it will flow out of the condenser against the pressure of the atmosphere.

219. Importance of Good Vacuum. With the advent of the steam turbine, the matter of high vacuum and good condensing apparatus has very greatly increased in importance. While the power of a compound

engine of the ordinary type will be increased only about four and one-half per cent, by increasing the vacuum from 26 to 28 inches, the power of a steam turbine for a given steam consumption will be increased in most cases by about $12\frac{1}{2}$ per cent. Increasing the vacuum from 28 inches to 29 inches will increase the power of steam turbine almost 10 per cent. Since the excellence of the vacuum obtained depends upon the efficiency of cooling and upon the efficiency of the air pump, it will be seen that the proper design of condensers and air pumps is a matter of very great economic importance in turbine installations.

220. Air in the Condenser. Water usually contains about 3 per cent of air, by volume, at ordinary temperatures, the volume of the air being estimated as **free air** (i.e., at atmospheric pressure and temperature). It may also contain a larger proportion by volume of carbon dioxide or ammonia, although it very seldom does. These gases may be dissolved in water, or they may be entrained (i.e., suspended in the water in the form of fine bubbles). All of these gases may be expelled by heating the water to boiling at atmospheric pressure in an open feed-water heater. Under the conditions ordinarily encountered in condenser practice, we may expect to have introduced into the surface condenser from the feed water about 1 cubic foot of free air for every 30 to 100 cubic feet of feed-water. The amount of air which enters the condenser on account of leakage when the exhaust piping and the rod packings are in good order, will be from 50 per cent to 150 per cent of that normally entering with the feed-water. The air-pump must therefore be designed to handle 1 cubic foot of free air for every 10 to 50 cubic feet of feed-water when it serves a surface condenser, or 1 cubic foot of free air for every 30 to 150 cubic feet of condensing water when it serves a jet or barometric condenser.

The pressure of the air in a surface condenser is usually about .30 to .50 pound per square inch absolute, when a first-class air-pump of usual proportions is employed. Since the pressure of the atmosphere is 14.7 pounds per square inch, it will be seen that the volume of the air in the condenser will be about 30 to 50 times its volume at atmospheric pressure. Consequently, if the usual percentage of air is associated with the feed-water, the volume of the air to be removed from a surface condenser will be roughly equal to the volume of the feed-water, and the air-pump must be proportioned accordingly. The pressure in the condenser will be equal to the pressure of the water vapor, which is determined by its temperature, plus the pressure of the air. It will be seen then, that if the pressure in the condenser is to be reduced, and a higher vacuum attained, we may do so either by reducing the temperature of the vapor by furnishing a larger quantity of cooling water, or else we may increase the capacity of the air-pump and so

reduce the pressure of the air. The pressure of the air is approximately inversely proportional to the capacity of the air-pump. In the case of a jet or barometric condenser, the volume of the air removed is about one-half the volume of the condensing water. Since, however, a condenser requires about 20 pounds of condensing water per pound of feed-water it will be seen that a jet or barometric condenser will need a much larger air-pump than will a surface condenser, to maintain the same vacuum.

It is possible by careful workmanship and proper design of the piping system, to make the exhaust pipe leading from the engine or turbine to the condenser, air tight. In the case of a properly designed turbine it is also possible to eliminate entirely all air leakage, although there will be some air leakage in the case of the best steam engines, around the valve stems and piston rods. In the case of a steam turbine plant it is possible, since no lubricating oil is carried into the condenser by the steam, to pump the condensed steam back into the boiler and use it over and over again. The feed-water obtained in this manner will, of course, be free from air, so that it is practicable in a first-class steam turbine plant to maintain a very high vacuum with a comparatively small air-pump when a surface condenser is used. When a jet condenser is used, the vacuum will not, of course, be as good as it would be with a surface condenser, unless the supply of condensing water is very limited. In case the supply of condensing water is limited, it will be found that a jet condenser will give a better vacuum, since in the case of the jet condenser, the condensing water is raised to the temperature of the vapor in the condenser, while in the case of a surface condenser there will necessarily be a difference between the temperature of the vapor and that of the discharged condensing water.

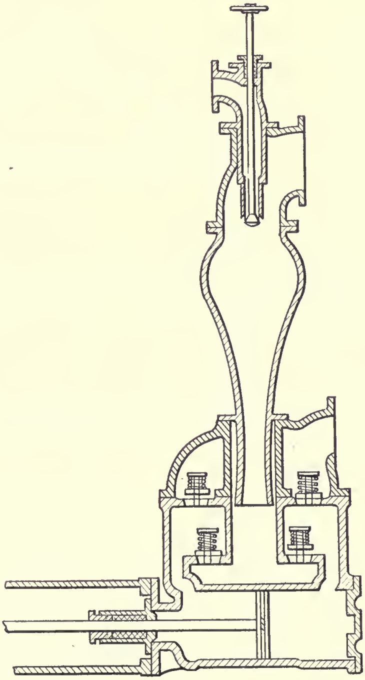


Fig. 112.—Section of a wet air pump.

221. The Air-pump. Air-pumps are of two classes, wet and dry pumps. The wet air pump removes the condensed steam or condensing water and the air together. A section of such a pump is shown in Fig. 112. Since the valves of the pump are so arranged that they are always

covered with water and the clearance space of the pump cylinder is filled with water at the end of the stroke, it will be seen that there will be no air in the clearance space when the suction stroke begins. This is a matter of great importance in air-pump design. If the clearance space of the air-pump is filled or partially filled with air at the end of the stroke, this air, if it is to be expelled at all, must be at atmospheric pressure. Consequently, during the suction stroke of the pump, this air will expand and prevent the pump from taking suction from the condenser during the greater part of the stroke, which will very greatly reduce the capacity of the pump for a given size of cylinder. By designing the pump so as to avoid the presence of air in the clearance space at the end of the stroke, this difficulty is avoided.

When a dry vacuum pump is used and the water is removed by a separate pump, it will be seen that it will be impossible to use this scheme in avoiding the presence of air in the clearance space. A dry vacuum pump may, however, be constructed with very little clearance, much less than is practicable in the case of a wet air pump. It may also run at a high speed, which greatly increases the capacity of a given size of cylinder. Different methods are adopted by different makers in order to avoid the reduction in capacity incident upon the clearance of the cylinder. One method is to use two cylinders, the larger one of which takes its suction from the condenser, and discharges into a receiver, while the second one takes its suction from this receiver and discharges into the air. It will be seen that the vacuum obtainable by this system of operation with pump cylinders of given clearance, is much greater than can be obtained by the use of a single cylinder of the same clearance, as may be seen from the following consideration. Assume a clearance of 5 per cent. Then with a single cylinder air-pump, the lowest pressure which can be reached will be about $^1/_{20}$ of an atmosphere when the quantity of air entering the condenser is negligible. This will be the lowest air pressure reached in the condenser in the case of a

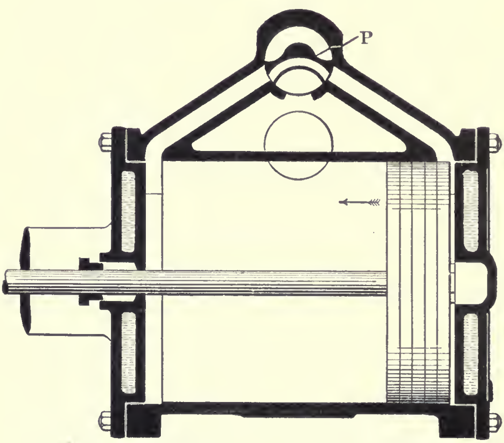


FIG. 113.—Section of a dry vacuum pump.

single cylinder air-pump, or in the receiver in the case of the compound air-pump. The extra cylinder of the compound air-pump will reduce the pressure in the condenser to $^1/_{20}$ of the pressure in the receiver, or $^1/_{400}$ of an atmosphere. If the quantity of air entering the condenser is appreciable, the air pressure will of course be greater in both cases than the amounts given.

A second method sometimes used is to connect the clearance space of the two ends of a double-acting cylinder by means of a valve which is opened for an instant after the contents of one end of the cylinder have been discharged against atmospheric pressure. A device of this kind is shown in Fig. 113. The port P connects the two ends of the cylinder during the instant just following the discharge of the contents of one end, and previous to the beginning of the compression of the contents of the other end. By this means, the pressure of the air in the clearance space is caused to fall to that of the opposite end of the cylinder, so that the card given by the air-pump has the form shown in full lines in Fig. 114, instead of that shown in dotted lines in the same figure, which would be the form of card given by the pump if it did not have the auxiliary port. The capacity of the pump without this auxiliary port would be proportional to the distance a–d. By means of this auxiliary port, the capacity is increased until it is proportional to the distance a–c. The efficiency of the pump when measured by the ratio between the work actually supplied to it and the work theoretically required to remove the air is less with the auxiliary port than without it. However, the advantage of very greatly increasing the capacity of the pump, without increasing the size of the cylinder, makes this a desirable principle of construction.

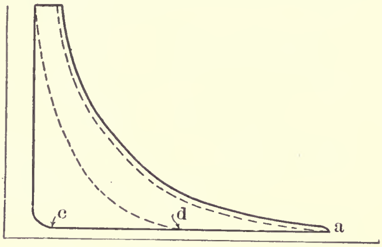


FIG. 114.—Card from the dry vacuum pump, shown in Fig. 113.

A third method of increasing the vacuum obtained by the use of a given air-pump is to force the air from the main condenser into a small auxiliary condenser by a blast of steam as shown in Fig. 115. In this figure A is the main condenser, B is a so-called augmentor condenser, and C is an aspirator operated by a steam blast. The purpose of the aspirator is to draw the air out of the main condenser, and to force it into the augmentor condenser in which its pressure will be materially greater than in the main condenser. The difficulty of removing the air from the augmentor condenser, will of course be very much less than of removing it from the main condenser in which the absolute pressure will be considerably lower. The steam used for the blast may be low

pressure steam, which has already been used in an engine or turbine and which has surrendered the most of its potential work.

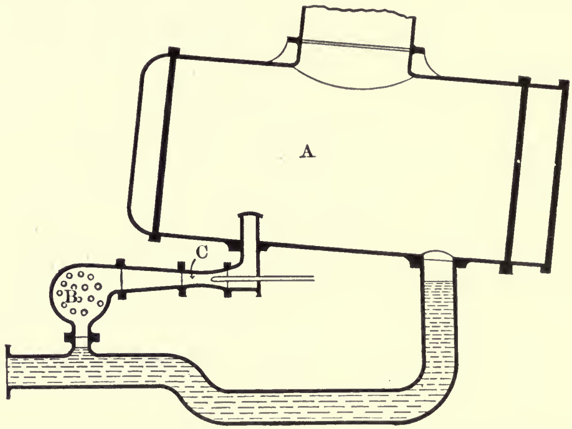


FIG. 115.—Section of a surface condenser equipped with an augmenter condenser.

PROBLEMS

1. Cooling water enters a surface condenser at a temperature of 56°. The temperature of the steam in the condenser is 90°. It takes four seconds for the water to pass through the condenser. The diameter of the condenser tubes is 1 in. What is the final temperature of the condensing water? Ans. 61.7° F.

2. A condenser having tubes 8 ft. long is arranged with four passes (i.e., so arranged that the water passes through the condenser four times, making its total travel 32 ft.). The tubes are $\frac{3}{4}$ in. diameter and the velocity of the water is $1\frac{1}{2}$ ft. per second. The temperature of the steam is 110° and of the entering water 80°. Find the temperature of the water discharged from the condenser. Ans. 101.92° F.

3. What quantity of cooling water enters each tube per second in problem 2? Ans. 0.288 lbs.

4. What quantity of heat is imparted by the condensing steam to this quantity of water? Ans. 6.31 B.T.U.

5. The condenser in Problem 2 is required to condense 30,000 lbs. of steam per hour, having a quality of 85 per cent. What quantity of heat must be absorbed by the cooling water? Ans. 26,250,000 B.T.U. per hr.

6. How many pounds of circulating water will be required to absorb this quantity of heat with the given rise in temperature? Ans. 333 lbs. per sec.

7. How many tubes will be required in each pass in order that this quantity of water may be circulated under the conditions in Problem 2? Ans. 1155.

8. What total area of cooling surface will the above number of tubes furnish? Ans. 7250 sq.ft.

9. How many pounds of wet steam will be condensed per square foot of cooling surface per hour? Ans. 4.14 lbs.

10. If the same number of tubes are arranged in a single pass, what will be the velocity and length of path of the circulating water? Ans. $\frac{3}{8}$ ft. per sec. and 8 ft.

11. What will be the final temperature of the circulating water?

Ans. Same as in Problem 2.

12. Why is the water usually circulated through a condenser with a high velocity in spite of the fact that the theoretical final temperature of the water is the same when it is circulated at low velocity?

13. Water is supplied to a jet condenser having a temperature of 60°. The temperature of the steam in the condenser is 110°, and the quality of the steam 90 per cent. How many pounds of circulating water must be supplied per pound of wet steam condensed? Ans. 18.5 lbs.

14. If the vacuum gage of the above condenser shows 24 inches, while the barometer shows 29 inches, what is the air pressure in the condenser?

Ans. 1.19 lbs. per sq.in.

15. What will be the reading of the vacuum gage of the above condenser, if the capacity of the air-pump is doubled? Ans. 25.2 in. Hg.

CHAPTER XIV

COMBUSTION

222. The Nature of Combustion Combustion, in the sense in which it is used in engineering, is the act of chemical union of the oxygen of the air with the carbon and hydrogen of a fuel, with an accompanying evolution of heat and light. The fuels commonly used for the purpose of steam generation are coal, coke, wood, and oils. These consist of carbon and hydrogen, together with traces of sulphur and phosphorus and also of oxygen and inert elements and compounds. Coal and coke contain free carbon. Coal, wood, and oils also contain compounds of carbon and hydrogen and of carbon, hydrogen and oxygen. Carbon and its compounds are the sources of practically all of the heat developed by combustion in thermodynamic apparatus.

223. Heat of Combustion. When a pound of any substance is burned in oxygen, we find that a definite quantity of heat is evolved as a result of the reaction. This heat is first imparted to the products of combustion, and by them conveyed to the bodies in their neighborhood. The **heat of combustion,** as this quantity is termed, is expressed in B.T.U. per pound of combustible. The heat of combustion of various substances is given in Table XII on page 215.

In the first column of this table will be found the names of the elements and chemical compounds, commonly found in fuels, and also of most of the common fuels. In the second column the physical state of the substances at atmospheric pressure and temperature is given. In the third column will be found the chemical symbol of the substance. In the fourth column will be found the atomic weight of the substance, in case it is an element. In the fifth column are given the molecular weights of elements and compounds. In the sixth column will be found the weight of the products of combustion in pure oxygen. In the seventh column are the chemical symbols of the products of combustion. In the eighth column is given the heat of combustion, assuming that the products of combustion are reduced to atmospheric pressure and temperature, and the steam formed is condensed to water. In the ninth column the latent heat at 70° F. of the steam formed by the combustion of 1 pound of the substance is given. In column ten is given the number of pounds of oxygen theoretically required per pound of combustible. In column

Table XII
PROPERTIES OF COMBUSTIBLE SUBSTANCES

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Name of Substance.	Physical State.	Chemical Symbol.	At. Wt.	Mol. Wt.	Products of Combustion.	Weight per Pound of Substance.	Heat of Combustion.	Latent Heat of Steam Formed.	Pounds O_2 per Pound of Substance.	Pounds Air per Pound of Substance.	Nitrogen per Pound of Substance.	Water Equiv. of Products of Combustion in O_2.	Water Equiv. of Products of Combustion in Air.
Carbon..........	Solid	C	12	...	CO_2 CO	3.67 2.33	14,500 4,300	....	2.67 1.33	11.50 5.78	8.83 4.45	0.693 0.566	2.848 1.651
Hydrogen........	Gas	H_2	1	2	H_2O	9.00	62,032	9471	8.00	34.80	26.80	4.15	10.69
Phosphorus......	Solid	P	31	...	P_2O_5	2.29	10,350	......	1.29	5.60	4.31	......	......
Sulphur..........	Solid	S	32.1	...	SO_2	2.00	4,030	......	1.00	4.35	3.35	0.308	1.125
Carbon monoxide.	Gas	CO	...	28	CO_2	1.57	4,380	......	0.57	2.48	1.91	0.297	0.763
Marsh gas........	Gas	CH_4	...	16	CO_2 H_2O	2.75 2.25	23,600	2370	4.00	17.40	13.40	1.556	1.883
Acetylene........	Gas	C_2H_2	...	26	CO_2 H_2O	3.39 0.69	21,430	727	3.08	13.40	10.32	0.960	1.212
Benzol...........	Vapor	C_6H_6	...	78	CO_2 H_2O	3.39 0.69	18,200	727	3.08	13.40	10.32	0.960	3.48
Ethylene.........	Gas	C_2H_4	...	28	CO_2 H_2O	3.14 1.29	21,350	1360	3.43	14.90	11.47	1.189	3.99
Alcohol..........	Liquid	C_2H_5OH	...	46	CO_2 H_2O	1.92 1.17	12,930	1233	2.09	9.10	7.01	0.903	2.61
Gasoline.........	Liquid	......	...	...	CO_2 H_2O	3.11 1.37	20,700	1440	3.48	15.10	11.66	1.221	4.062
Fuel oil..........	Liquid	......	...	...	CO_2 H_2O	3.06 1.32	20,000	1390	3.41	14.80	11.42	1.187	3.972
Sugar...........	Solid	$C_{12}H_{22}O_{11}$	...	342	CO_2 H_2O	1.55 0.578	7123.6	608	1.13	4.92	3.78	0.559	1.489
Cellulose *.......	Solid	$C_6H_{10}O_5$	...	162	CO_2 H_2O	1.63 0.556	7,470	585	1.185	5.16	3.97	0.564	1.533
Pennsylvania anthracite †..		9.2% ash			CO_2	....	12,900	per lb. of dry coal					
Pennsyl. bituminous coal †..		7.9% ash			CO_2 & H_2O	....	13,940						
Colorado lignite †		13.1% ash			CO_2 & H_2O	....	10,700						

* The combustible matter of wood consists of cellulose and turpentine. The heating value of air-dried wood varies from 6600 to 8000 B.T.U. per pound.
† The results given are the average for the region.

eleven will be found the number of pounds of air containing this quantity of oxygen, and in column twelve the number of pounds of nitrogen contained in this quantity of air. In column thirteen will be found the water equivalent of the products of combustion in pure oxygen, and in column fourteen the water equivalent of the products of combustion in air. The **water equivalent** of a body may be defined as its heat absorbing capacity in B.T.U. per degree rise in temperature, and may be found by multiplying its mass in pounds by its specific heat. The method of computing the quantities in the table may be inferred from the following paragraphs on the combustion of carbon and hydrogen.

It is sometimes convenient in making computations on combustion to make use of the **heat of formation** of a compound. This may be defined as the quantity of heat evolved in forming a pound of the substance in question, by the process of combustion. The heat of formation of the common products of combustion will be found in Table XIII, together with their specific heats (**or water equivalents per pound**).

TABLE XIII

PROPERTIES OF THE CONSTITUENTS OF FLUE GASES

Name of Substance.	Symbol.	Heat of Formation.	Weight of O in One Pound.	Specific Heat at Constant Pressure.
Carbon monoxide	CO	1845	0.571	0.243
Carbon dioxide	CO_2	3950	0.727	0.189
Water	H_2O	6890	0.890	0.461
Sulphur dioxide	SO_2	2010	0.499	0.154
Phosphorus pentoxide	P_2O_5	4520	0.563	
Oxygen	O_2			0.217
Nitrogen	N_2			0.244
Air				0.237

224. The Combustion of Carbon. The simplest case of combustion is that of pure carbon. When incandescent carbon in small lumps is placed in a draft of air, the carbon unites with the oxygen of the air to form carbon dioxide and carbon monoxide. If the temperature is high and the air supply is scanty, large quantities of CO will be formed. If the gases of combustion are subsequently brought into contact with air while sufficiently hot, the CO will burn to CO_2, evolving a large amount of heat. If, however, the gases of combustion are permitted to cool before bringing them in contact with the air, the CO will not burn.

Dry air consists of 20.7 per cent by volume of oxygen, and 79.3 per cent by volume of nitrogen and other inert gases. The inert gases will be termed nitrogen in this discussion. By weight air consists of 23 per

cent of oxygen and 77 per cent nitrogen. When a pound of carbon is burned to CO_2 in air, we have for the formula of the reaction.

$$\underset{12}{C}+\underset{32}{O_2}=\underset{44}{CO_2}.$$

Underneath the chemical symbols are written the molecular weights of the substances. These indicate that 12 pounds of carbon unite with 32 pounds of oxygen to form 44 pounds of carbon dioxide. By simple proportion it will be seen that 1 pound of carbon unites with 2.67 pounds of oxygen to form 3.67 pounds of carbon dioxide. Since air contains 23 per cent of oxygen by weight, it requires 11.6 pounds of air to supply this 2.67 pounds of oxygen. If, however, we attempt to burn a pound of carbon with just sufficient air to complete the combustion, we will find that some carbon monoxide will be formed and some of the oxygen will fail to combine, and that it is necessary to supply an excess of air in order to insure the complete combustion of the carbon and obtain its full heating value.

225. The Combustion of Carbon in Air. It may be shown, however,[1] that the economical use of fuel requires that the amount of the excess air be made as small as possible. In practice it is found that the fuel is utilized to the best advantage when from 30 to 50 per cent excess of air is supplied. Assume that 15 pounds of air are used to burn 1 pound of carbon. In this 15 pounds of air are 3.45 pounds of oxygen and 11.56 pounds of nitrogen. As a result of the combustion $2\frac{2}{3}$ pounds of the oxygen unite with 1 pound of carbon to form $3\frac{2}{3}$ pounds of CO_2 and 0.78 pounds of oxygen remain in the free state, as does also the 11.55 pounds of nitrogen. The products of combustion are usually termed **flue gas**. Since the volume of the carbon dioxide is the same as the volume of the oxygen from which it was formed, the volume of the 16 pounds of flue gas is the same as the volume of the 16 pounds of air at the same temperature.

The heat generated by the above combustion will be found by careful calorimeter measurements to be 14,500. B.T.U. The water equivalent of the products of combustion is found by multiplying the weight of each constituent by its specific heat at constant pressure (since the combustion occurs at constant pressure), and adding the products. The water equivalent of the CO_2 is 3.67×0.217; of the oxygen, 0.78×0.217; of the nitrogen, 11.55×0.244. Adding the products, it will be found that the water equivalent of the 16 pounds of gas is 3.792. A quantity of very great importance in the theory of fuel economy is the **theoretical temperature of the fire,** which is found by adding to the initial temperature of the air and fuel which may be taken as the temperature of the

[1] See Arts. 246 and 248.

fire room, the rise in temperature theoretically produced by the heat of combustion. Dividing the heat of combustion by the water equivalent of the flue gases, we have for the theoretical rise in temperature due to the combustion, 3830° F., and assuming that the original temperature of the air and the carbon was 70°, we have for the theoretical temperature of the fire 3900° F.[1]

226. The Combustion of Hydrogen. The combustion of hydrogen is a reaction represented by the formula:

$$\begin{array}{ccc} 2H_2 + & O_2 = & 2H_2O. \\ 2\times 2 & 32 & 2\times 18 \end{array}$$

From the molecular weights it will be seen that 1 pound of hydrogen requires 8 pounds of oxygen for its combustion, the result being 9 pounds of water vapor. Assuming that 50 per cent excess of air is used, it will be seen that this air must contain 12 pounds of oxygen and that the total quantity of air will be 52.2 pounds. The heat of combustion will be 62,032 B.T.U

The products of combustion will, of course, be 9 pounds of water, 4 pounds of oxygen, and 42.2 pounds of nitrogen. The water equivalent of these products of combustion will be 15.00. In reducing the products of combustion from the temperature of the fire to their original temperature, the steam formed will be reduced to water. The latent heat of evaporation of this steam, which will be $9\times 1051.5 = 13{,}640$ B.T.U. at 70° F., will be liberated by this condensation. Since this heat is latent, it has no effect in raising the temperature of the products of combustion. Hence, the heat available for producing the rise in temperature is $63{,}032 - 13{,}640 = 49{,}392$ B.T.U. This quantity is sometimes known as the lesser heat of combustion of hydrogen. Dividing this quantity by the water equivalent, we have for the theoretical rise in temperature 3290° and for the theoretical temperature of the furnace 3360°.

[1] In practice, the temperature of the fire, when 1 pound of carbon is burned by 15 pounds of air, is very much less than the figure given, for several reasons. In the first place, the fire radiates heat into the boiler and the walls of the furnace, and the heat available for producing the rise in temperature is diminished by the amount so radiated. In the second place, when the temperature has risen to between 3000 and 3200° F., carbon monoxide is formed instead of carbon dioxide, and the quantity of heat evolved is less than 14,500 B.T.U. per pound of carbon. This carbon monoxide, if mixed with sufficient air, will of course burn to carbon dioxide as soon as the temperature falls sufficiently. This phenomenon is known as suppressed combustion. In the third place, it is found that the specific heat of a gas increases with the temperature, so that even if complete combustion did occur, and no heat was radiated, the temperature realized would not be the theoretical temperature obtained by assuming a constant specific heat. However, so long as the actual temperature of the furnace is sufficiently great to maintain rapid combustion, it is of no practical importance what the actual temperature is. On the other hand, the theoretical temperature of the fire is of the greatest importance, since it determines the efficiency of the boiler plant.

227. The Combustion of Chemical Compounds. When a compound of carbon and hydrogen is burned, the carbon burns to carbon dioxide and the hydrogen to water. In estimating the quantity of oxygen required, we must first find the weight of carbon and of hydrogen in 1 pound of the substance and allow sufficient oxygen to burn both the carbon and the hydrogen. The weight of the products of combustion can be computed from the weight of carbon and hydrogen present in a pound of the substance. The theoretical rise in temperature may be computed by subtracting from the heat of combustion the latent heat of evaporation at atmospheric temperature of the water formed by the combustion, and dividing the difference by the water equivalent of the products of combustion.

Some fuels, as for instance wood, which consists largely of a chemical substance termed cellulose, contain oxygen as well as carbon and hydrogen. The chemical reaction resulting from the combustion of cellulose may be written

$$\underset{162}{C_6H_{10}O_5} + \underset{6\times 32}{6\ O_2} = \underset{6\times 44}{6\ CO_2} + \underset{5\times 18}{5\ H_2O}.$$

From this it will be seen that 1 pound of cellulose consists of .444 pounds of carbon, 0.0618 pounds of hydrogen, and 0.494 pounds of oxygen. The oxygen is, of course, not a combustible, and the hydrogen is already united with the oxygen in the compound, and hence is not available as a combustible. The only combustible present is the .444 pounds of carbon whose combustion will, in theory, yield 6240 B.T.U. In any chemical compound containing oxygen, the oxygen is united with one of the combustible elements, and the heat of combustion is usually approximately equal to that of the combustible substances not united with oxygen present in 1 pound of the compound.

It may be noted that the heat of combustion of a compound may be more or less than the heat of combustion of the elements forming it. For instance, the heat of combustion of acetylene is found to be 21,429 B.T.U. One pound of acetylene consists of $^{12}/_{13}$ of a pound of carbon and $^{1}/_{13}$ of a pound of hydrogen. The heat of combustion of the carbon will be $^{12}/_{13} \times 14{,}500 = 13{,}400$ B.T.U., and of the hydrogen $63{,}032 \times {}^{1}/_{13} =$ 4850 B.T.U. The sum of these, or 18,250 B.T.U., is less than the heat of combustion of acetylene. A chemical compound must be decomposed before it can be burned. The decomposition of the compound absorbs heat in many cases, for instance in the case of marsh-gas. Such a compound is termed **endothermic.** In other cases, acetylene for instance, heat is given up as a result of the decomposition. Such a compound is termed **exothermic.** The heat of combustion of exothermic substances is more, and of endothermic substances less, than the heat of combustion of their chemical constituents.

228. Flue Gas Analysis by the Orsat Apparatus. In engineering work computations relative to the efficiency of combustion must usually be based upon the results of a chemical analysis of the flue gases. Flue gases are usually analyzed by means of the Orsat apparatus, which determines the volume of CO_2, of O_2, and of CO present in a sample of dry gas. The remainder of the gas is nitrogen and other inert elements. The form of the Orsat apparatus is shown in Fig. 116. The gas to be analyzed is drawn into the measuring burette *A* by manipulating the water bottle *B*. The measuring burette holds exactly 100 c.c. of gas. By manipulating the water bottle, this gas is first passed into the treating pipette *C*, containing a solution of potassium hydroxide, which absorbs the CO_2. After being passed several times into pipette *C*, the gas is withdrawn and measured in the burette *A*. The shrinkage in volume indicates the per cent by volume of CO_2. In like manner, the gas is introduced into the pipette *D* containing a solution of potassium pyrogallate, which absorbs the oxygen, and then into pipette *E* containing an acidulated solution of cuprous chloride, which absorbs the carbon monoxide. The shrinkage in volume in each case indicates the per cent by volume which the gas removed bears to the whole quantity taken. Since the gas is in contact with water while it is being measured, and the apparatus is kept at constant temperature, the gas is always saturated with water vapor, and the pressure of the water vapor is constant. Consequently, when the volume of the gas in the apparatus is reduced by absorbing a given proportion of it, the same proportion of the total quantity of the water vapor present is condensed, and although the gas comes from the stack loaded with moisture and is analyzed while it contains this moisture, the results given by the apparatus are those which would be obtained by analyzing a sample of the gas after the moisture has been abstracted.

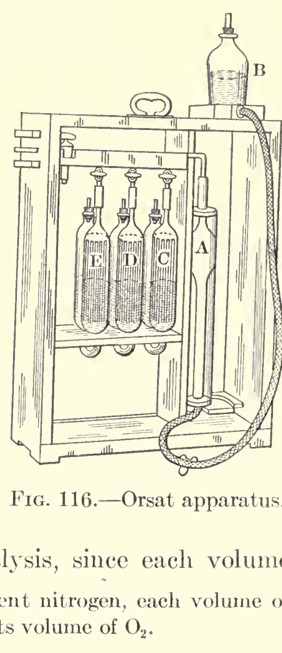


FIG. 116.—Orsat apparatus.

When a substance containing hydrogen is burned, and the flue gases analyzed by the Orsat apparatus, it will be found that the volume of the oxygen accounted for by the analysis will be less than 26.1 per cent of the volume of the nitrogen.[1] The per cent of oxygen accounted for by the analysis is equal to the per cent of CO_2 plus the per cent of O_2 plus one-half the per cent of CO shown by the analysis, since each volume

[1] Since air consists of 20.7 per cent O_2 and 79.3 per cent nitrogen, each volume of atmospheric nitrogen will be mixed with 26.1 per cent of its volume of O_2.

of CO_2 requires one volume of O_2 and each volume of CO one-half a volume of O_2 for its formation.

229. Computations of the Efficiency of Combustion Based on a Volumetric Analysis of the Flue Gases. Let N equal the number of cubic centimeters of nitrogen, O the number of cubic centimeters of free oxygen, Coo the number of cubic centimeters of carbon dioxide, and Co the number of cubic centimeters of carbon monoxide in each 100 c.c. of dry flue gas, as shown by the flue gas analysis. Then, for the number of cubic centimeters of oxygen accounted for in 100 c.c. of flue gas, we will have

$$\left(O + Coo + \frac{Co}{2}\right) \text{ c.c.} \quad \ldots \ldots \quad (1)$$

The oxygen represented by the nitrogen in 100 c.c. of flue gas will be

$$0.261\ N \text{ c.c.} \quad \ldots \ldots \quad (2)$$

The difference will give the number of cubic centimeters of oxygen represented by the nitrogen in 100 c.c. of flue gas which combined with hydrogen to form water vapor. This will be

$$0.261\ N - \left(O + Coo + \frac{Co}{2}\right) = A \text{ c.c.} \quad \ldots \ldots \quad (3)$$

Assume for a unit of mass 1 c.c. of a gas whose molecular weight is 1. Then the weight of the carbon in 1 c.c. of CO_2 or CO will be 12, since the atomic weight of carbon is 12, and the weight of the carbon in 100 c.c. of flue gas will be

$$(Coo + Co)\ 12.$$

The number of cubic centimeters of oxygen accounted for in 100 c.c. of flue gas is .261 N. The weight of 1 c.c. of oxygen will be 32. Consequently the weight of oxygen accounted for in 100 c.c. of flue gas will be

$$0.261\ N \times 32.$$

The number of pounds of carbon burned by each pound of oxygen supplied will therefore be

$$\frac{(Coo + Co)\ 12}{0.261\ N \times 32} = \text{C};$$

reducing, this becomes

$$\text{C} = \frac{1.44\ (Coo + Co)}{N} \text{ lbs.} \quad \ldots \ldots \quad (4)$$

In like manner it may be shown that the number of pounds of hydrogen burned per pound of oxygen supplied will be

$$\mathrm{H} = \frac{0.48\,A}{N}\ \text{lbs.} \qquad (5)$$

The excess of air supplied will be

$$\frac{383\,O}{N - 3.83\,O}\ \text{per cent.} \qquad (6)$$

of that theoretically necessary.

Adding C from (4), and H from (5), we will have the number of pounds of combustible per pound of oxygen supplied. For the number of pounds of air supplied per pound of combustible we will have

$$\frac{4.35}{\mathrm{C}+\mathrm{H}}. \qquad (7)$$

For the number of pounds of nitrogen in the flue gas per pound of combustible, we will have

$$\frac{3.35}{\mathrm{C}+\mathrm{H}}.$$

For the number of pounds of carbon dioxide in the flue gas per pound of combustible we will have

$$\frac{5.27\,Coo}{N\,(\mathrm{C}+\mathrm{H})}\ \text{lbs.} \qquad (9)$$

For the number of pounds of carbon monoxide in the flue gas per pound of combustible, we will have

$$\frac{3.36\,Co}{N\,(\mathrm{C}+\mathrm{H})}\ \text{lbs.} \qquad (10)$$

For the number of pounds of water vapor in the flue gas per pound of combustible, we will have

$$\frac{4.20\,A}{N\,(\mathrm{C}+\mathrm{H})}\ \text{lbs.} \qquad (11)$$

For the number of pounds of free oxygen in the flue gas per pound of combustible, we will have

$$\frac{3.83\,\mathrm{O}}{N\,(\mathrm{C}+\mathrm{H})}\ \text{lbs.} \qquad (12)$$

The heat evolved per pound of combustible will be

$$\frac{6340\ Co+20900\ Coo+29400\ A}{N\ (\mathrm{C+H})}\ \text{B.T.U.} \quad . \quad . \quad . \quad . \quad . \qquad (13)$$

Dividing this by the water equivalent, which may be readily obtained from the weights of the various constituents of the flue gas per pound of combustible, we will obtain the theoretical rise in temperature. Adding the air temperature, we will obtain the theoretical fire temperature. For an example of the application of the theory of combustion to an actual boiler test the reader is referred to Art. 252.

230. The Composition of Coal. The fuel of greatest practical value in engineering work is coal. Coal is fossil vegetable matter which under the combined influence of heat and pressure has been changed materially in its chemical and physical form. It consists of carbon and hydro-carbon compounds, together with varying amounts of water and ash. The moisture, which varies from 0.25 to 8.0 per cent by weight of the coal, may be driven off by heating it to a temperature of 220° F. for an hour or more. If the coal be heated to a red heat, and the air is excluded during the process, the hydro-carbons will be driven off, and carbon and ash will remain behind. The hydro-carbons so driven off are termed volatile matter. The substance remaining is termed coke, and the carbon of the coke is termed fixed carbon. If coke is heated in the presence of the air, for a sufficient length of time it will burn, leaving behind the ash, which usually consists of sand, clay, iron oxide, lime, etc. When the moisture, volatile matter, fixed carbon, and ash are determined, the process is termed proximate analysis.

Coal is classified as bituminous, semi-bituminous and anthracite, according as it contains more or less volatile matter. Bituminous coals are coals containing from 20 to 50 per cent of volatile matter. They may be divided into coking and non-coking coals. If the volatile matter of a coal will melt before vaporizing the coal is said to **coke,** or **cake,** since it will be fused into a compact mass of coke while burning. If, on the other hand, the volatile matter vaporizes without melting, the coal is termed non-coking coal. Semi-bituminous coals are those containing from 12 to 20 per cent of volatile matter. Anthracite coals are those containing less than 12 per cent of volatile matter. The volatile matter of coal consists mostly of hydro-carbon compounds which may, and in the case of bituminous coals high in volatile matter, usually do contain some oxygen.

231. The Heating Value of Coal. The heating value of coal varies greatly, as may be inferred from the great variation in composition. The moisture and ash, of course, have no heating value. The oxygen contained in the hydrogen compounds also has no heating value, and

furthermore, since it is already combined with a portion of the hydrogen of these compounds, it reduces the heating value of the combustible elements present. The heating value of the coal may be obtained approximately from the formula

$$\text{B.T.U.} = 14{,}500\,C + 63{,}000\left(H - \frac{O}{8}\right) + 4{,}100\,S,$$

in which C is the number of pounds of carbon, H is the number of pounds of hydrogen, and S is the number of pounds of sulphur per pound of coal, as obtained from the ultimate chemical analysis.

A more exact result for the heating value may be obtained by the use of a calorimeter. The form of calorimeter invented by Parr is especially suitable for engineering use and gives dependable results when calibrated by burning in it chemically pure sugar of known heating value.

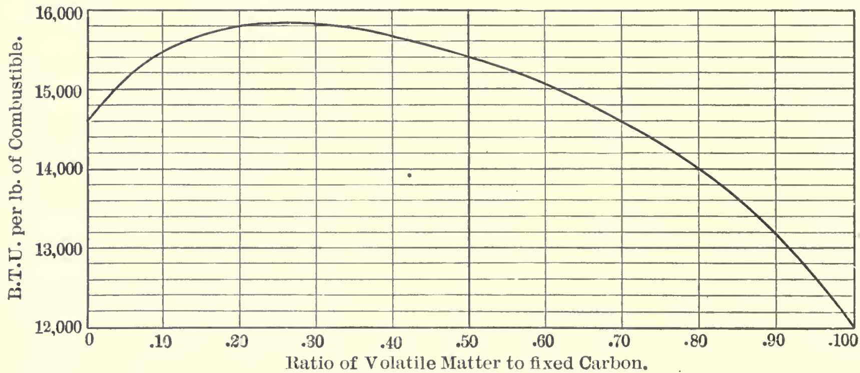


FIG. 117.—The heating value of coal.

The heating value of coal may also be determined with considerable accuracy from the proximate analysis as follows: Deduct the percentage of moisture and ash as shown by the proximate analysis, the result will be the percentage of combustible in the coal. Divide the per cent of volatile matter by the per cent of fixed carbon. From the curve in Fig. 117, obtain the heating value per pound of combustible, of coal having this ratio of volatile matter to fixed carbon. Multiplying this value by the per cent of combustible in the coal will give the heating value of the coal in B.T.U. per pound.

232. The Combustion of Coal. Coal is burned on a grate whose purpose it is to admit air to the under side of the fire and to permit of the removal of ash from the fire. Grates are of various forms and are arranged in various ways in order to adapt them to different kinds of coal and conditions of service. The simplest grate consists of parallel bars of cast

iron upon which the fire is laid. Assuming such a grate, with a fire built upon it, we may note the following phenomena with respect to the combustion of coal:

Pure carbon burns without flame to carbon dioxide. In the presence of an excess of carbon, this carbon dioxide is decomposed, forming carbon monoxide, which burns with a flame. A fire of pure carbon, or coke, of moderate depth will therefore burn with little or no flame. In order to utilize the fuel as economically as possible, it is necessary, as has already been noted, to burn the carbon with the least possible excess of air. The excess of air will be determined by the intensity of the draft and the thickness of the fire, consequently it is necessary to so adjust the thickness of the fire that the quantity of air will be the minimum necessary to maintain the required rate of combustion. If the quantity of air admitted to the under side of the fire be less than that required for complete combustion, some air must be admitted at the top of the fire, in order to burn the carbon monoxide which will be formed. It is usual in practice to carry the fire slightly thicker than is necessary and then to admit a proper quantity of air over the fire in order to secure complete combustion.

233. The Production of Smoke. If upon such a coke fire a layer of bituminous coal is spread, the water and volatile matter in the coal will be quickly vaporized by the heat. A considerable proportion of this volatile matter is tar. Tar vapor is difficult to burn, since its kindling temperature is high, and the rate of combustion, on account of the comparatively great density of the vapor, is much lower than is the case with other combustible gases. As the tarry vapors rise from the coal, they are mingled with the products of combustion and with an excess of free oxygen. If the temperature of the mass is above the kindling point of the tar vapor, and it remains at this temperature for a sufficient length of time to permit of a complete mixture of the various gases, the tar will be burned to water and carbon dioxide. If, however, the vapor is cooled below its kindling point before it has an opportunity to completely mix with the air, it will not become ignited, but will be discharged from the chimney in the form of unburned vapor. As soon as the temperature of the chimney gas becomes low enough to permit of the liquefaction of these unburned vapors, they are condensed upon particles of dust and form specks of soot, which give to the gases coming from the chimney the intensely black appearance which we note in soft coal smoke.

The amount of potential heat carried away in the flue gases as a result of such incomplete combustion is usually comparatively small, often being less than two per cent of the total heat of the coal, when the smoke is very dense and black. Most of the volatile matter is burned in the furnace, and only a small portion of it escapes combustion and

produces the smoke. However, the pollution of the air of our cities by vast quantities of black smoke is highly objectionable for many reasons. Hence many attempts have been made to compel owners of steam plants to so construct and operate their furnaces as to avoid the production of smoke. It will be apparent that there are two possible methods of preventing the escape of smoke from a chimney. The first one is to cool the products of combustion and mechanically extract the dust and tar from them. This method is commercially impracticable, although it has been seriously proposed as a remedy for the smoke nuisance. The second method consists in thoroughly mixing the vapors coming from the fire

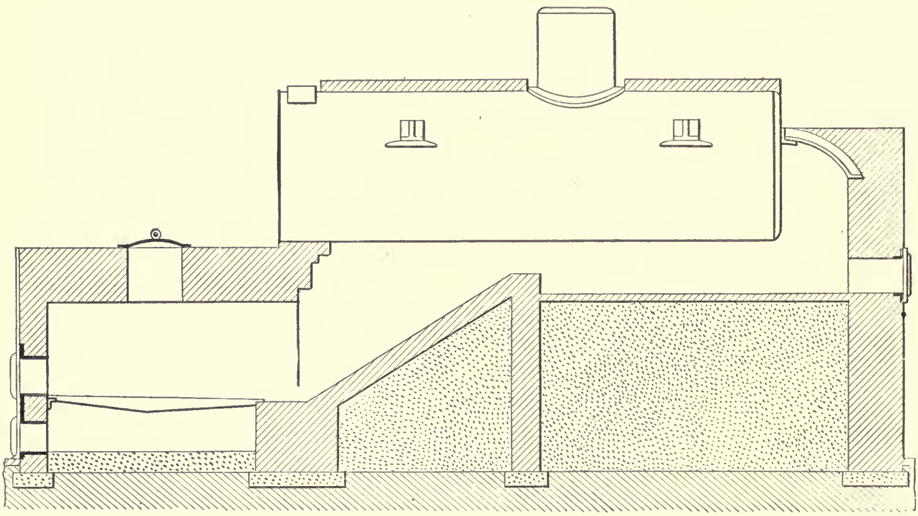


FIG. 118.—Horizontal return tubular boiler equipped with a Dutch oven furnace.

with a sufficient quantity of highly heated air before they have been cooled below their kindling point.

When a combustible gas or vapor is intimately mixed by diffusion with an excess of highly heated air, combustion is very rapid, only a fraction of a second being necessary to practically complete the reaction. However, in the case of a coal fire, hand fed in the ordinary manner, the gases are not thoroughly mixed as they rise from the fire, since the ari tends to pass through the holes of the fire, and the volatile matter of the coal is separated from the surrounding air by the products of combustion. Columns of air arise from some portions, and columns of combustible vapors from other portions, of such a fire. These tend to pass through the furnace in parallel streams, diffusion and combustion occurring only at the boundaries of the streams. As soon as the streams of gas encounter

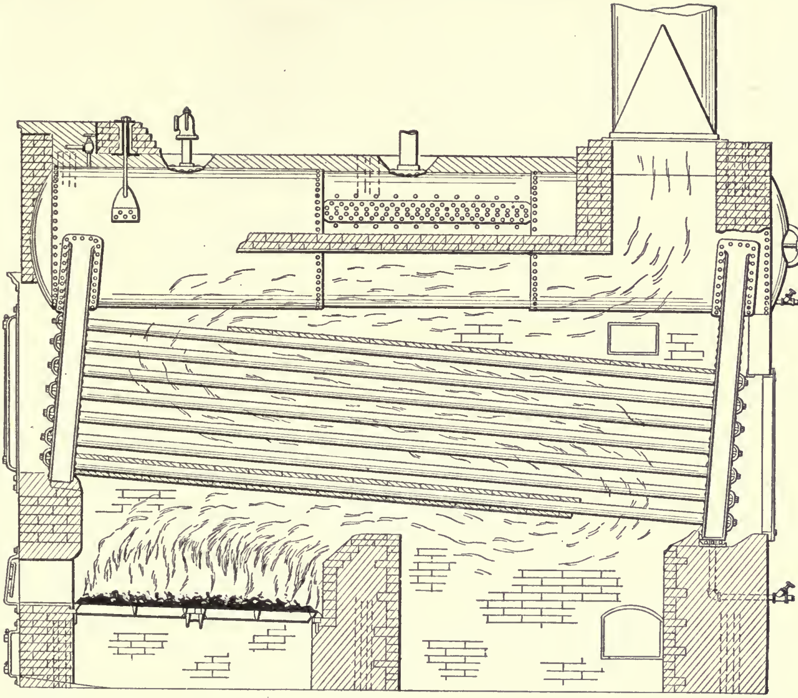


FIG. 119.—Furnace equipped with tile roof attached to lower row of boiler tubes.

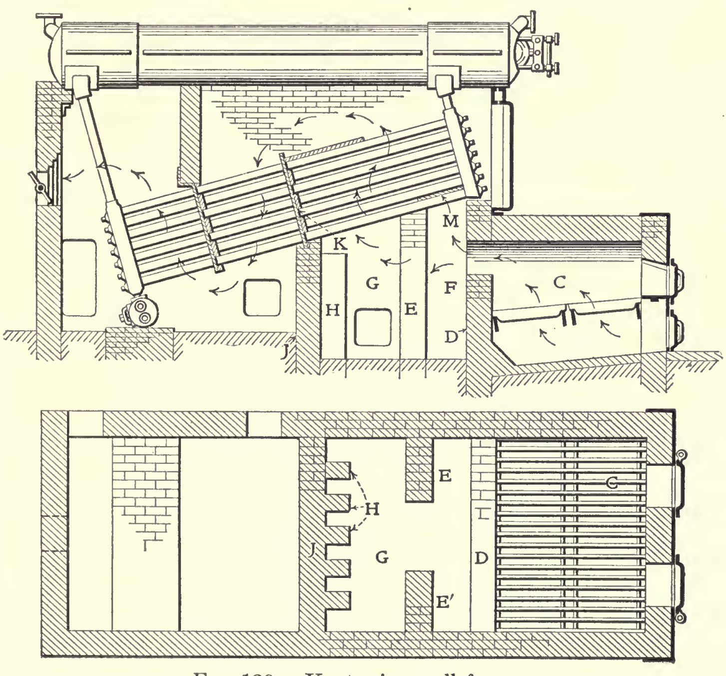


FIG. 120.—Kent wing-wall furnace

a cold metallic surface they are cooled below their kindling point, combustion ceases, and smoke is produced.

234. Methods of Smoke Prevention. Three general methods are in use for securing a thorough mixing of the combustible vapors with heated air before they are cooled. The first of these methods consists in leading the combustible gases and the air coming from the fire into a chamber of sufficient size, so that they may remain in it long enough to be thoroughly mixed. In order that the temperature of this chamber may be above the kindling point of the gases, it is lined with fire brick, which soon becomes incandescent. In order to make the process of mixing quicker and more thorough this chamber is sometimes so arranged that the currents of gas and air are baffled in their progress and made to mix as a result of the eddying produced by the baffles.

235. The Dutch Oven. The Dutch oven furnace illustrated in Fig. 118 is an example of a combustion chamber of incandescent fire brick in which the gases are permitted to mix. It consists of a large chamber over the fire with walls and roof of fire brick. If the condition of the fire is kept uniform by careful firing, the gases will be quickly and thoroughly mixed in this chamber and combustion will be complete. The same result is accomplished by the furnace shown in Fig. 119, in which the roof of the furnace is not a brick arch, but consists of tiles of fire clay attached to the lower row of tubes of the boiler. The Kent wing-wall furnace illustrated in Fig. 120 is an example of a furnace in which the gases are caused to mingle by means of baffles. As the gases pass from the Dutch oven over the bridge wall D, they strike against the wing walls E, whose purpose it is to cause the streams of gas to mingle as a result of the eddy currents which they create.

236. The Mechanical Stoker. The second method of securing smokeless combustion consists in introducing the coal continuously into the furnace and in so directing the gases rising from the fire that they shall be caused to mingle. The chain-grate stoker and the rocking-grate stoker are examples of this method of securing smokeless combustion. A chain-grate stoker is illustrated in Fig. 121. Coal is fed into the hopper and the grate consists of a series of parallel bars attached at either end to a chain. The motion of the chains, which are operated by gearing, carries the coal bodily into the fire. A rocking-grate stoker is shown in Fig. 122. The coal passes from the hopper on to the grate, which is inclined, and the rocking motion of the grate causes it to slide down, burning as it goes. The chain-grate stoker is particularly adapted to the use of anthracite coal and non-coking coals. The rocking-grate stoker is particularly adapted to the use of coking coals.

Since coal is introduced continuously into the fire when a stoker is used, and large quantities of volatile matter are not given off at once,

as is usually the case with poor hand firing, the gases have a much better opportunity to mix with a sufficient quantity of heated air, and combustion will accordingly be more complete. A chain grate or rocking grate stoker, however, will not give absolutely smokeless combustion unless a fire-brick combustion chamber of one of the types described in Art. 235 is used in connection with it.

237. The Down-draft Furnace. The third method of securing smokeless combustion consists in passing the combustible gases through the fire. This is accomplished either by drawing the air and products of combustion downward through the fire, or by introducing the fresh coal

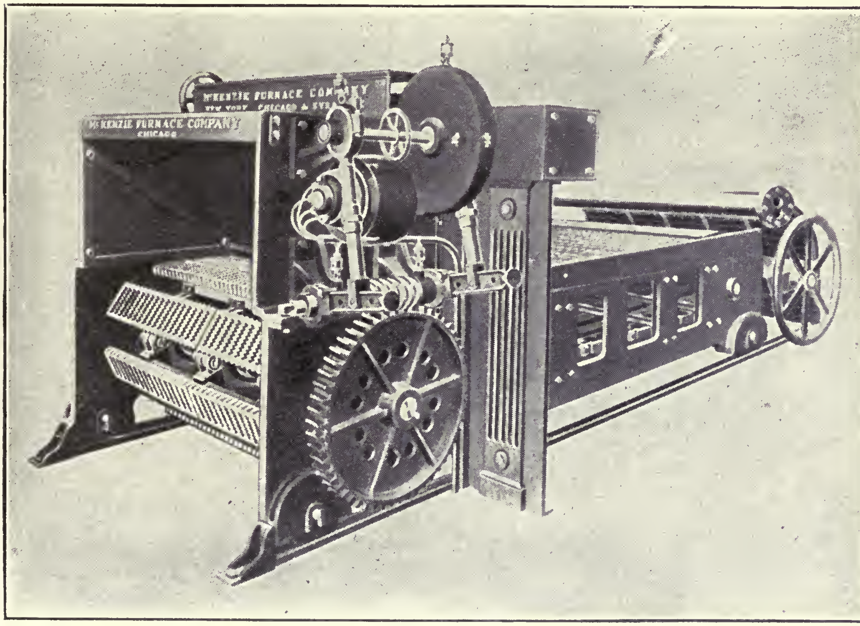


Fig. 121.—Chain grate stoker.

at the bottom of the fire. The first of these methods is the one employed in the Hawley Down-draft furnace illustrated in Fig. 123. The fire is built upon the upper grates, which are iron tubes filled with water from the boiler. The fresh coal is spread on top of the fire and the air passes downward through the fire. The temperature of the incandescent coke through which the air and volatile matter passes is sufficient to decompose the tarry vapors, and insure their complete combustion. As the coke is formed, it drops through onto the lower grate, where it is completely burned

The under-feed stoker illustrated in Fig. 124 accomplishes the same result by the second method. The fresh coal is introduced at the bottom

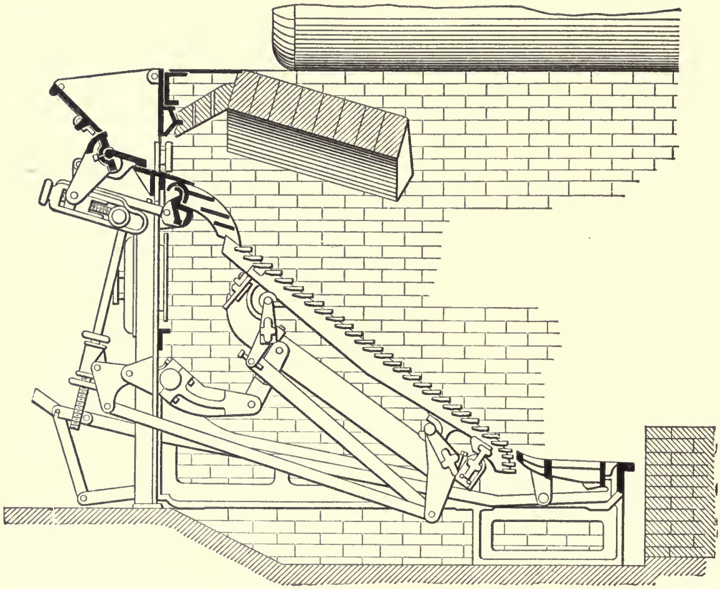


FIG. 122.—Rocking-grate stoker.

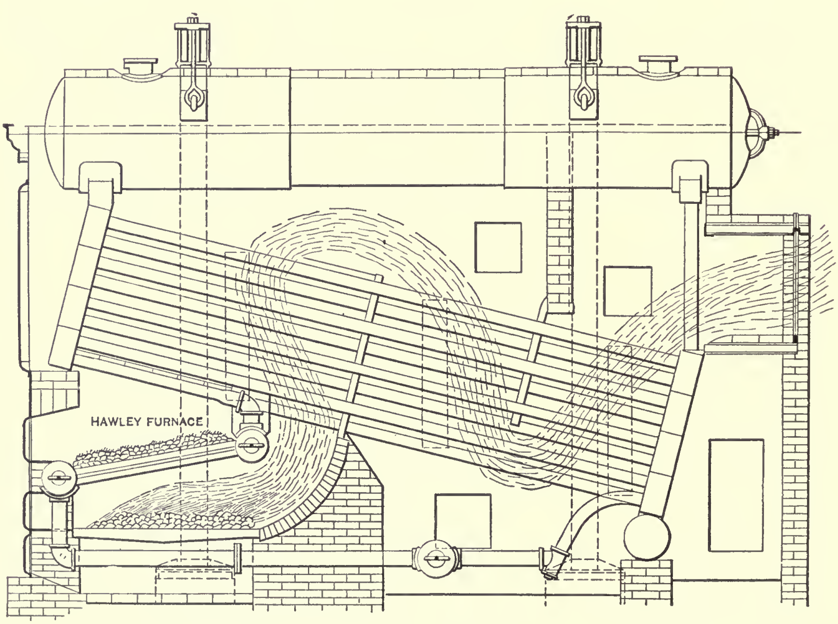


FIG. 123.—Water tube boiler equipped with Hawley down-draft furnace.

of the fire by means of the conveyor. The gases distilled from it pass upward through the bed of incandescent coke, where they are thoroughly mixed with highly heated air and burned. A stoker of this type gives absolutely smokeless combustion, but is only adapted for coals which have comparatively little ash which does not "clinker."

238. The Practical Management of Fires. Returning to a consideration of the hand-fired furnace described in Art. 232, it may be noted that the ash contained in the coal gathers upon the grates as the fire burns and must be removed at intervals by **cleaning the fire,** i.e., by separating the ash from the coke, hoeing out the ash, and then spreading the coke again over the grates. Some coals contain very little ash, so that there is no difficulty in its disposal. Other varieties of coal, however, contain large amounts of ash, and sometimes the ash is of such a character that it is melted by the heat of the fire, forming masses of glass termed **clinkers.** Anthracite coal almost invariably produces ash which clinkers when the

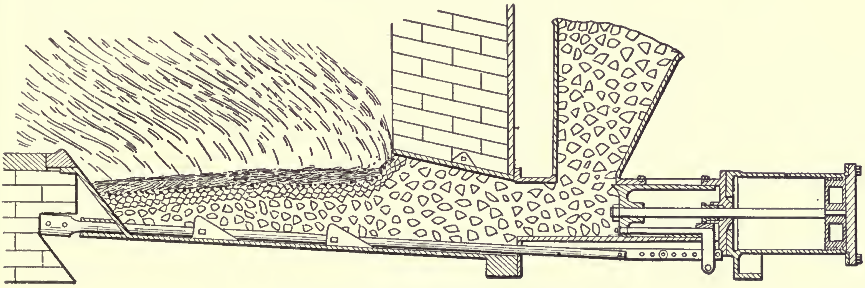


Fig. 124.—Underfeed stoker.

fire is very hot. In order to avoid the production of clinkers with anthracite coal, it is customary to introduce steam into the ash pit. The steam is decomposed by the carbon, forming hydrogen and carbon monoxide. The reaction absorbs heat and cools the under side of the fire below the melting point of the ash. The gases formed are subsequently burned when they reach the upper side of the fire, and they there liberate the heat which they absorb from the lower layer of fuel.

When the fuel used is a coking coal, it will be found that the coal fuses together into a compact mass which does not permit of the passage of air through the fire. In order to allow the air free access to all parts of the fire, it is necessary to **bar** or **slice** the fire, or in other words, to break up the mass of coke by the use of a huge poker. The finest sizes of anthracite coal are in certain parts of the country a very cheap fuel. They are difficult to burn, however, since if the openings in the grate are of sufficient size to allow a proper air supply, much of the coal will fall through. In order to use such fuel, it may be mixed with a coking coal in the propor-

tion of three parts of anthracite to one of bituminous coal. The coking of the mixture cements the anthracite particles together, so that they are much more readily handled in the furnace and the loss of coal through the grates is very much reduced.

The proper handling of a coal fire is a matter of great importance in the economical operation of a boiler plant. It is necessary to have sufficient draft to operate the fire at the required rate of combustion. The fire must be kept of uniform thickness and the coal must be spread evenly upon all parts of it unless the "coking method" of firing is used. If a coking coal is used, the fire must be barred as soon as the coke is formed. The thickness of the fire must be adjusted to the rate of combustion, so that the least possible excess of air is used. The fire doors must be kept closed as much as possible, since when they are open large volumes of cold air will be drawn into the furnace and heat will be wasted in warming this air to the temperature of the chimney gas.

In the **coking method** of firing the fresh coal is placed at the very front of the fire, next to the fire doors. As soon as this coal is coked, it is pushed back over the grates. The vapors rising from it while it is coking pass backward over the bed of incandescent coke which forms the remainder of the fire, and are there burned. In the **alternate method** of firing, first one side and then the other is covered with coal. The gases rising from the fresh coal are mingled in the combustion chamber with the excess of air coming from the other side of the fire, and there burned.

PROBLEMS

1. One pound of carbon is burned with 20 lbs. of air. Find the water equivalent of the flue gases. Ans. 4.86.

2. Find the theoretical rise in temperature. Ans. 2980°.

3. One pound hydrogen is burned with 80 lbs. of air. Find the water equivalent of the flue gases. Ans. 20.4.

4. Find the theoretical temperature of the fire, assuming the original temperature to be 100°. Ans. 2680° F.

5. One pound of acetylene is burned with 40 lbs. of air. Find the water equivalent of the products of combustion. Ans. 9.78.

6. Find the rise in temperature resulting. Ans. 2190° F.

7. The Orsat analysis of a flue gas shows 10 per cent of CO_2, 9 per cent of O_2 and 1 per cent of CO_2. How many c.c. of oxygen is represented by the nitrogen in 100 c.c. of flue gas? Ans. 20.9 c.c.

8. How many c.c. of this oxygen united with hydrogen? Ans. 1.4 c.c.

9. How many pounds of carbon were burned per pound of oxygen supplied? Ans. 0.198 lbs.

10. How many pounds of hydrogen were burned per pound of oxygen supplied? Ans. 0.0084 lbs.

11. What per cent excess of air was supplied? Ans. 75.8%

12. How many pounds of air were supplied per pound of combustible? Ans. 21.1 lbs.

13. How many pounds of carbon monoxide are there in the flue gas per pound of combustible? Ans. 0.204 lbs.

14. How many pounds of carbon dioxide are there in the flue gas per pound of combustible? Ans. 3.20 lbs.

15. How many pounds of water vapor are there in the flue gas per pound of combustible? Ans. 0.356 lbs.

16. How many pounds of free oxygen are there in the flue gas per pound of combustible? Ans. 2.09 lbs.

17. What is the heat evolved per pound of combustible? Ans. 15,550 B.T.U.

18. What is the water equivalent of the products of combustion per pound combustible? Ans. 5.23.

19. What is the theoretical fire temperature, assuming the initial air temperature to be 70°. Ans. 3050° F.

20. A coal on analysis is found to consist of 4 per cent water, 10 per cent ash, 20 per cent volatile matter, and 66 per cent fixed carbon. Find its heating value. Ans. 13,600 B.T.U.

CHAPTER XV

THE STEAM BOILER

239. The Horizontal Return Tubular Boiler. The **steam boiler** is a metallic vessel in which steam is generated under pressure, by the application of heat. The essential parts of a steam boiler are the the furnace, the boiler proper, and the setting. In addition, in order to operate it, every boiler must be provided with certain auxiliaries such as a chimney, a feed pump, a pressure gage, a water column, and suitable piping, valves, etc. The sectional view in Fig. 125 shows a

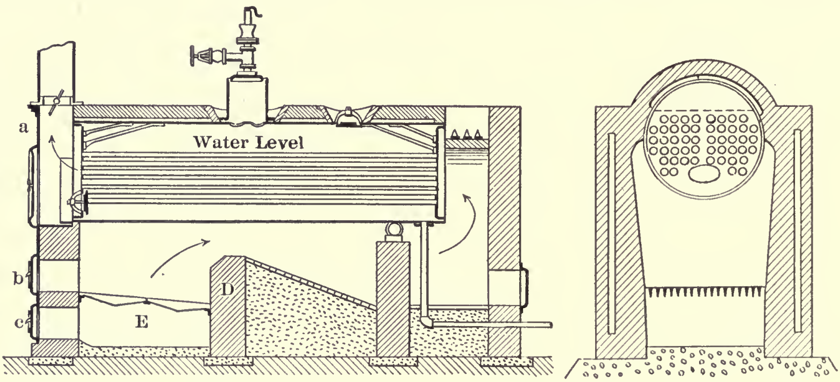


FIG. 125.—Longitudinal and transverse sections of a horizontal return tubular boiler and setting.

standard type of boiler, known as the **horizontal return tubular boiler.** In this figure, *a* is the boiler front, which is usually made of cast iron. In this front are three sets of doors, set *b* being termed the **fire doors,** and set *c* the **ash-pit doors,** while the upper doors are known as **clean-out doors.** The clean-out doors allow access to the front end of the boiler, for cleaning. *D* is a brick partition termed the **bridge wall.** Between the fire doors and the **bridge wall** are placed the grates *E,* upon which the fire is built. Back of the bridge wall is a space termed the **combustion chamber.** The boiler proper consists of a cylindrical drum with flat ends, which is made by rolling together plates of steel from $^1/_4$ to $^5/_8$ of an inch in thickness, and riveting to the ends of the hollow cylinder, so formed, circular plates termed **tube sheets.** Extending

from the front to the rear tube sheets are rows of tubes, about three or four inches in diameter, which carry the gases of combustion through the water space. After passing along the under side of the boiler and through the combustion chamber, the hot gases from the fire enter the tubes and pass forward to the breeching or gas passages which serve to carry them to the chimney. The tubes, as will be seen from the transverse section of the boiler and setting, occupy the lower two-thirds of the boiler, while the upper one-third is reserved as steam space. The boiler is filled with water to a depth of about three inches above the tops of the tubes. When the boiler is in operation, this water

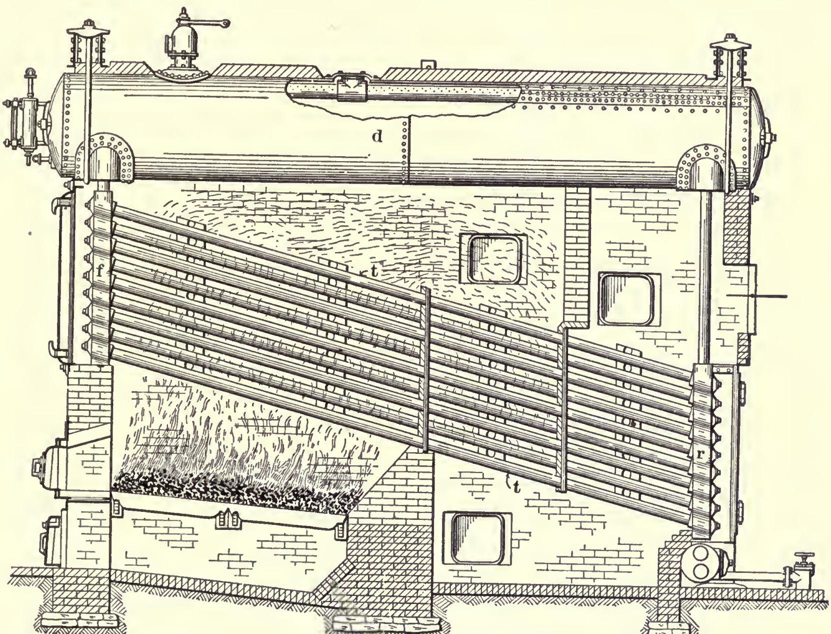


Fig. 126.—Longitudinal section of a horizontal water tube boiler.

has the temperature of vaporization corresponding to the pressure of the steam, usually from 300° to 400° F. The temperature of the metal of the boiler is only a little higher, so that when the hot gases are led along the under side of the shell, and back through the tubes, they are promptly cooled, surrendering their heat to the water, and evaporating it.

240. The Horizontal Water Tube Boiler. Fig. 126 shows another type of boiler which is known as a **horizontal water tube boiler.** This differs from the former type in that the water is contained in a number of tubes about which the fire and hot gases are caused to play. These

tubes pass from the **rear headers** to the **front headers,** and their contents are caused to circulate by passing forward through tubes *t–t*, upward through the front headers *f*, rearward through the drum *d*, and downward through the rear headers *r*. Since the rear headers contain water only, while the front headers contain a considerable amount of steam

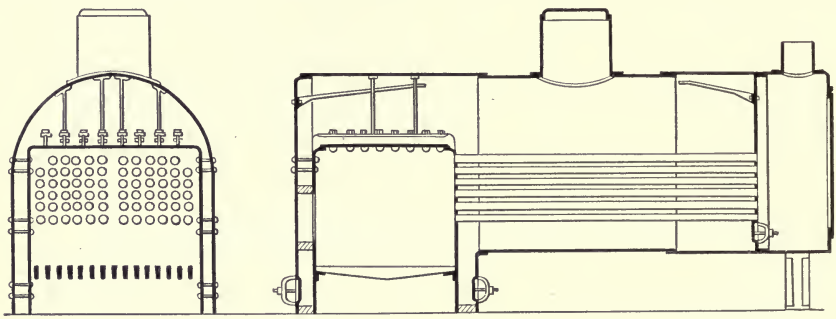


FIG. 127.—Transverse and longitudinal sections of a locomotive type boiler.

in the form of foam and bubbles, the resulting difference in density causes this circulation to go on with a considerable velocity.

241. Classification of Boilers. Various other types of boilers are in use, but they all consist of a furnace, either of firebrick, or of metal surrounded by water, of heating surfaces, usually in the form of tubes

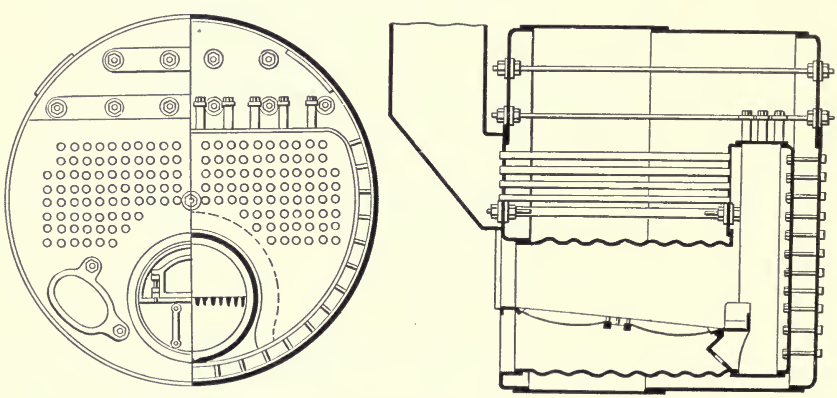
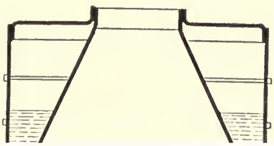
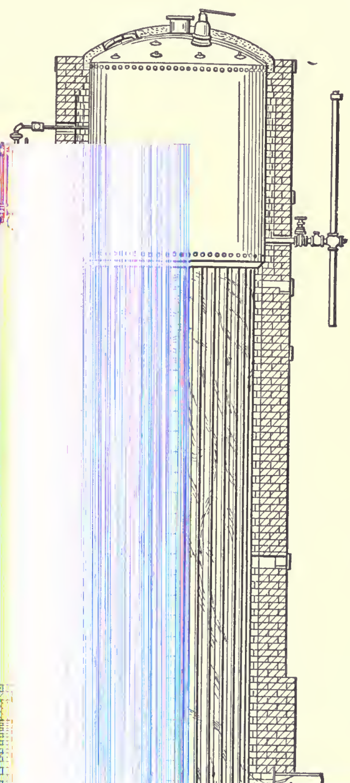


FIG. 128.—Transverse and longitudinal sections of a single furnace Scotch marine boiler.

surrounded by, or containing water, and of gas passages which bring the furnace gases into intimate contact with the heating surfaces. They also contain a drum, or steam storage reservoir, whose function it is to separate the water from the steam by gravity, so that dry steam may be drawn from the boiler.

The boilers ordinarily in use may be divided into three classes. Boilers of the first class are known as **fire tube boilers.** In such boilers the flames from the furnace are caused to pass through tubes which are





as **water tube boilers.** In such boilers the heating surface consists of tubes surrounded by hot gases and containing water. The horizontal water tube boiler already described is of this type. Sometimes the tubes are in a vertical position, as in the boiler illustrated in Fig. 130, which is termed a **vertical water tube boiler,** and sometimes curved tubes are used, as in the boiler illustrated in Fig. 131. The third type

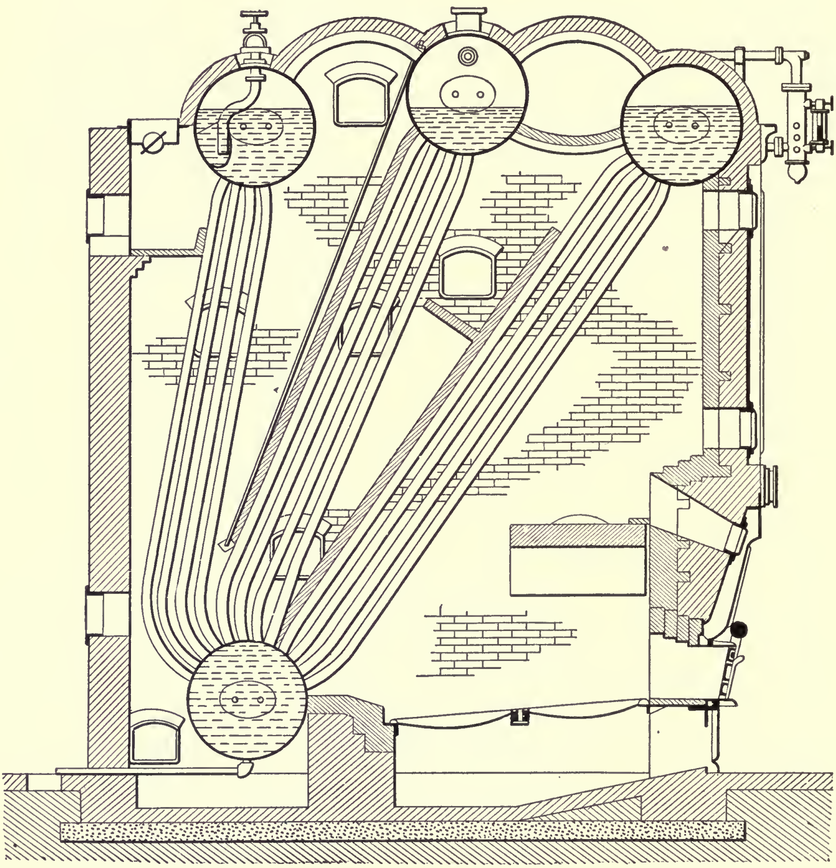


FIG. 131.—Section of a Stirling water tube boiler.

of boiler is known as a **flash boiler.** In this boiler the feed-water which is to be vaporized flows into one end of a long tube which is heated, and the water is vaporized before it has passed through the tube. Such boilers, being light for their power, are often used in steam-driven automobiles. The Parker steam generator, illustrated in Fig. 132, is an illustration of such a boiler adapted for stationary service. In all there

are several hundred types of boilers in use, but they are all modifications or combinations of the three types mentioned.

242. Theory of the Steam Boiler. When the gases resulting from combustion leave the fire they have a high temperature. As they pass through the boiler, encountering metallic surfaces which are in contact with hot water, they are reduced in temperature. Finally, after passing through the boiler, they are discharged into the chimney at a temperature very much less than the temperature of the fire, but still considerably higher than the temperature of the water in the boiler. The rate at which they will impart heat to any surface with which they are in

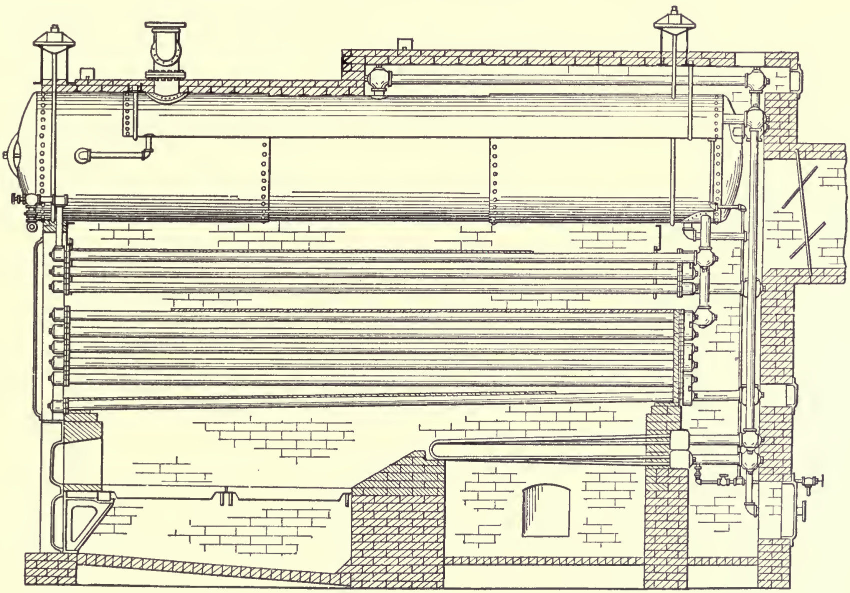


Fig. 132.—Section of a Parker down-flow boiler with superheater.

contact is proportional to the difference in temperature between the gases and the surfaces,[1] so that they lose heat the most rapidly to the

[1] The assumption sometimes made, that the rate of heat transfer is proportional to the **square** of the temperature difference, is untenable. The resistance encountered by the heat in its passage from the gas to the water may be divided into three parts. The first and largest part is the thermal resistance of the layer of cool gases which is in immediate contact with the heating surface, and which is prevented by friction from having the motion of the main body of gas. The second part is the resistance of the metal plates, which is very small. The third part is the resistance of a thin layer of steam which separates the metal plates from the water. Each of the resistances must follow the usual law of heat conduction, which is that the rate of heat conduction is proportional to the conductivity (or inversely proportional to the specific resistance) of the material, proportional to the temperature difference, and

first surface with which they come in contact. In order to determine the rate of heat conduction, and the final temperature of the gases leaving a boiler, we will assume that a quantity of hot gas is flowing through a tube surrounded by water; that the initial temperature of the gas as it comes from the fire is T_f; that the diameter of the tube is 3.82 inches, so that the surface exposed to the action of the gas per foot of length of the tube is one square foot; that the gas, when in contact with the tube, will impart to it H heat units per square foot per hour for each degree difference in temperature between the gas and the tube; that the temperature of the inner surface of the tube T_w is sensibly the same as the temperature of the water; and that W pounds of gas pass each cross-section of the tube per hour. If the gases flow through a certain section of the tube whose length is dL, and the difference in temperature between the gases passing this section and the tube itself is T, they will be reduced in temperature while passing the section by the amount dT. The quantity of heat imparted to the water through the walls of this section of the tube will be $HTdL$ heat units per hour. The heat lost by the gas passing this section in one hour will be equal to the change in temperature dT, times the specific heat at constant pressure, times the weight of gas passing the cross-section each hour. Writing these quantities equal we will have

$$T\,H\,dL = dT\,C_p\,W. \qquad (1)$$

Assuming the length of the tube to any point to be L feet, we will have, since dT is negative,

$$\int_T^{T_f - T_w} \frac{dT}{T} = \frac{H}{C_p\,W}\int_0^L dL. \qquad (2)$$

Integrating this we have

$$\log_e \frac{T_f - T_w}{T} = \frac{H\,L}{C_p\,W}. \qquad (3)$$

Clearing we have

$$\log_e (T_f - T_w) = \log_e T + \frac{H\,L}{C_p\,W}. \qquad (4)$$

Solving for T

$$\log_e T = \log_e (T_f - T_w) - \frac{H\,L}{C_p\,W}. \qquad (5)$$

inversely proportional to the thickness of the material. This law has been thoroughly established by experiment, and any other assumptions will be found to lead to results which are mutually inconsistent. Experiments which have tended to establish the idea that the rate of heat transmission is proportional to the square of the temperature difference have invariably been so conducted as not to eliminate the effects or radiation.

We may write this, for simplicity,

$$\log T = A - \frac{B\,L}{W}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

in which

$$A = \log (T_f - T_w) \qquad \text{and} \qquad B = .43\frac{H}{C_p}, \quad . \quad . \quad . \quad . \quad (7)$$

and the logarithms are common, and not natural logarithms.

243. The Efficiency of the Heating Surface. The principal item in the cost of boiler operation is the cost of the fuel burned. It is therefore highly desirable to use the least possible quantity of fuel in generating a given quantity of steam. To do this, we must of course transfer to the water in the boiler the greatest possible proportion of the heat generated by the combustion of the fuel in the furnace. If we disregard (as is permissible in most practical cases) the latent heat of the water vapor formed by combustion, we find that the heat generated in the furnace is equal to the theoretical rise in temperature of the products of combustion, multiplied by the water equivalent of the flue gas. The theoretical rise in temperature is of course equal to the difference between the theoretical fire temperature T_f, and the temperature of the fire room, T_a. The amount of heat lost to the chimney is equal to the difference between the temperature of the flue gas T_c and the temperature of the fire room T_a, multiplied by the water equivalent of the flue gas. The heat utilized by the boiler will then be equal to the difference between the theoretical fire temperature and the temperature of the flue gas, multiplied by the water equivalent of the flue gas. Since the temperature of the flue gas is equal to the temperature of the water in the boiler plus the final difference in temperature between the water and the gas, we may write

$$T_c = T_w + T. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

In order to find the **theoretical efficiency of the heating surface** of a boiler, we must divide the heat transferred to the water by the heat generated by the combustion, in which case we will obtain the equation

$$E = \frac{T_f - T_c}{T_f - T_a} = \frac{T_f - T_w - T}{T_f - T_a}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

An inspection of the above equation will show that the efficiency of the heating surface may be increased by increasing the theoretical fire temperature, since such an increase adds the same quantity to both numerator and denominator of the fraction. The efficiency of the heating surface may also be increased by reducing the temperature difference between the water and the gases leaving the boiler, and by increasing the temperature of the air supplied to the fire.

244. Nominal Power of a Boiler. The rate of driving of a boiler may be defined as the quantity of heat transferred from the hot gas to the water per square foot of heating surface per hour. When the percentage of radiation loss is small, this is very nearly, although not exactly, proportional to the **rate of evaporation,** or the number of pounds of water actually evaporated per square foot of heating surface per hour. In discussing the theoretical efficiency of boilers, in this chapter the rate of driving will be assumed to be proportional to the rate of evaporation. Usual practice allows about 10 to 12 square feet of heating surface per nominal **boiler horse-power.** A boiler horse-power is the capacity to evaporate 34.5 pounds of water per hour from and at 212°, which requires the transmission of 33,480 B.T.U. per hour from the gas to the water. It will be seen that at the usual rating, the heating surface of a boiler is required to transmit, on the average, about 3000 B.T.U. per square foot per hour. This may be considered the normal rate of driving, and other rates of driving will be expressed as a per cent of the normal rate.

An increase in the quantity of coal burned on the grate of a boiler furnace will, of course, increase the quantity of heat supplied to the boiler in a given time. The rate of heat supply is proportional to the **rate of combustion,** which may be defined as the number of pounds of coal burned per square foot of grate per hour. The number of pounds of gas per hour passing through the boiler (the quantity W in equation (5), Art. 242) is proportional to the rate of combustion, and may be found by multiplying the rate of combustion by one plus the number of pounds of air per pound of fuel, and the product by the grate area. The rate of driving of a given boiler is proportional to the product of the rate of combustion into the efficiency of the boiler at that rate of driving.

245. Temperature of the Flue Gas. Referring again to equation (5), Art. 242, we may, since the heating surface in square feet is equal to the length of the tube in feet, write it in the form

$$\log T = \log (T_f - T_w) - \frac{2\,H\,F}{W},$$

in which T is the temperature difference between the water in the boiler and the gases leaving the boiler, T_f is the temperature of the water in the boiler, H is the number of heat units transmitted per hour, per square foot of heating surface per degree difference in temperature, from the gas to the water, F is the number of square feet of heating surface in the boiler, W the number of pounds of gas passing through the boiler per hour, and the logarithms are common and not natural logarithms. Plotting the relation between the amount of heating sur-

face and the final temperature of the gases for a constant rate of combustion, we will have the curve illustrated in Fig. 133.

An inspection of this curve shows that the temperature of the gas flowing through the tube continually approaches, but never reaches, the temperature of the water, T_w (i.e., T approaches zero). We will also find that while the reduction in temperature is rapid at first, it finally becomes slow and that further increase of the heating surface does not produce any material reduction in the temperature of the gas leaving the boiler. We may therefore conclude the following in regard to the efficiency of the heating surface of a boiler: First, the portion of the chimney loss which is due to the difference in temperature between T_f and T_w, may be reduced to any desired extent by sufficiently increasing the heating surface of the boiler. Second, there is a practical limit to the desirable extension of the heating surface, since if this surface is made too great, the amount of heat transferred through that portion of the surface last encountered by the gas will be too small to be of practical use. Third, the higher the initial temperature of the gases (i.e., the greater the temperature of the fire), the greater the proportion of the total heat generated in the furnace which may be transferred to the water. Fourth, the less the weight of gases passing through the boiler each second (i.e., the less the rate of combustion), the greater the proportion of their total heat which will be transferred to the water. Fifth, the greater the conductivity of the heating surface (i.e., the greater the value of H), the greater the efficiency of the boiler.

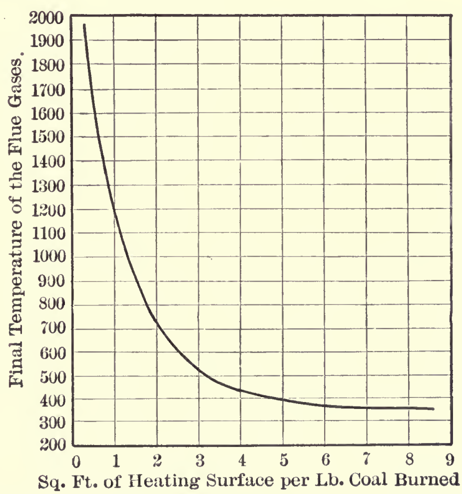


FIG. 133.—Relation between the rate of combustion and the temperature of the flue gas.

246. Conditions of Maximum Boiler Efficiency. It will thus be seen that in order to operate a boiler at maximum efficiency, it is necessary that the quantity of air supplied per pound of fuel shall be a minimum, since a low ratio of air to fuel will give a high furnace temperature, and also will reduce the weight of gases passing through the boiler in a given time. An appreciable increase in the conductivity of the heating surfaces can only be obtained by reducing the thickness of the film of cold

gases in contact with the heating surface. This may be done by arranging the gas passages of the boiler in such a way that the streams of gas shall be thoroughly broken up and intermingled at every possible point, thus bringing fresh supplies of hot gas into contact with the heating surface. It has been suggested that this will be best accomplished by drawing the gases through the boiler at high velocity and reducing the area of the gas passages. It has also been suggested that an excess of air be deliberately added to the gases in order to increase their velocity, or that a portion of the flue gases be re-circulated for the same reason. When these methods are tested in practice, however, they prove to be defective. In case the passages are greatly restricted and a power-driven fan is employed to draw the air through at high velocity, it is found that the value of the power taken by the fan exceeds the value of the increased efficiency of the boiler, so that the system is commercially less efficient than the common method of boiler operation. If the quantity of gases is increased, the loss of efficiency due to the extra weight of these gases circulated is greater than the gain realized by the more thorough contact secured (i.e., W in the equation in Art. 245 increases as fast as, or faster than H, as might be expected). None of these schemes are practically as satisfactory as a further addition of heating surface would be in securing an increase in efficiency, and the addition of an economizer as described in Chapter XVI, or an air preheater, is even more satisfactory.

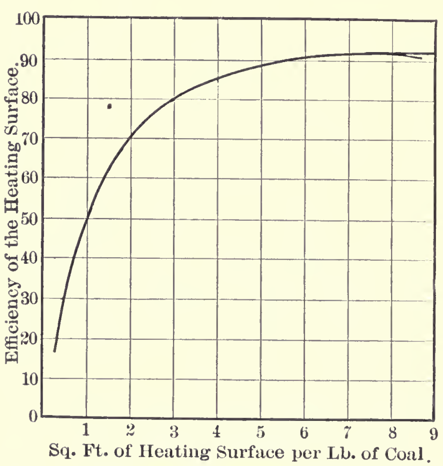


FIG. 134.—Relation between the efficiency and the rate of combustion.

247. Effect of Rate of Driving on the Efficiency of the Heating Surface. Substituting the value of T as obtained from Fig. 133, in equation (2), Art. 243, and solving for the efficiency, we obtain the curve given in Fig. 134, which gives us the relation between the efficiency of the heating surface and the amount of heating surface for a constant rate of combustion. This same curve also, of course, shows the relation between the efficiency of the heating surface and the rate of combustion when the amount of heating surface remains constant. It will be seen that, as the heating surface is increased, or the rate of combustion reduced, the efficiency of the heating surface increases slowly, approaching, but

never equaling the value

$$E = \frac{T_f - T_w}{T_f - T_a}.$$

The curve in Fig. 135 shows the relation between the rate of driving and the efficiency of the heating surface. Inspection shows that the efficiency falls off more rapidly for a given percentage increase in the rate of driving, than for the same percentage increase in the rate of combustion.

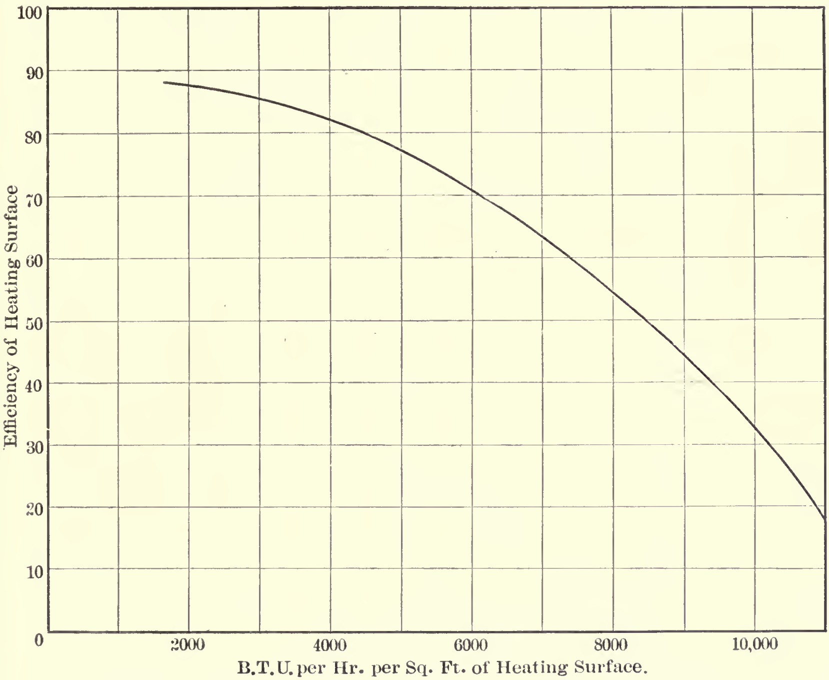


FIG. 135.—Relation between efficiency and rate of driving.

248. Effect of Air Leakage. The effect of an increase in the number of pounds of air per pound of fuel upon the efficiency of a boiler is shown in the two curves in Fig. 136. The dotted curve shows the relation when the rate of combustion is constant, while the full line shows the relation when the rate of driving is constant. It will be noted that an increase in the ratio of air to fuel has a more serious effect in reducing the efficiency of the boiler than any of the other elements affecting this efficiency, for the range of values commonly found in practice.

249. Effect of Increasing the Conductivity of the Shell. An increase in the conductivity of the boiler plates is equivalent to an extension of the boiler surface. This explains why the removal of scale from a

boiler does not very greatly increase the efficiency of a boiler, although it may considerably increase the conductivity of the heating surface. At the normal rates of driving even a considerable reduction in the resistance of the heating surface to the passage of heat, has very little effect upon the efficiency of the boiler, just as at the normal rate of driving a considerable increase in the heating surface will have but

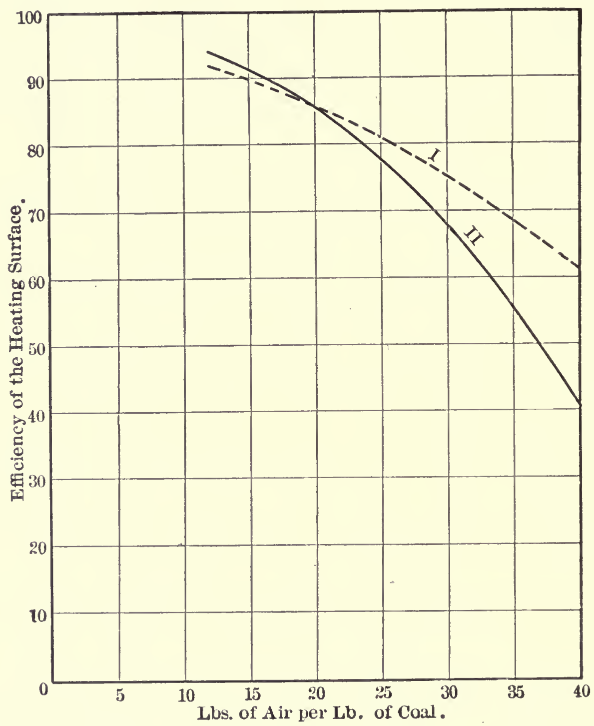


Fig. 136.—Relation between the efficiency and the ratio of air to fuel.

Curve I is for a constant rate of combustion.
Curve II is for a constant rate of driving.

little effect upon the efficiency. An increase in the pressure of the steam, and therefore of the temperature of the water in the boiler, has but little effect upon the efficiency of the boiler, as may be seen by reference to Fig. 137, which shows the relation between the temperature of the steam and the efficiency of the boiler, for the conditions given.

250. Effect of Radiation on Boiler Efficiency. A boiler, like any other heated body, radiates a considerable amount of heat into the surrounding air. This amount may vary from 2 to 20 per cent of the quantity of heat generated in the furnace and depends upon the temperature of the boiler and furnace, and the thoroughness with which the

boiler is clothed in non-conducting materials and the area of radiating surface exposed. This loss by radiation is independent, or almost so, of the rate of driving.

In considering the efficiency of the boiler, it is necessary to consider the loss due to radiation. The surface of the setting increases with the square of the dimensions of the boiler, while the heating surface of the boiler increases with the cube of its dimensions, hence the radiating surface increases in proportion to the two-thirds power of the heating surface. The loss of heat due to radiation may be taken as being independent of the rate of driving and proportional to the radiating surface, or

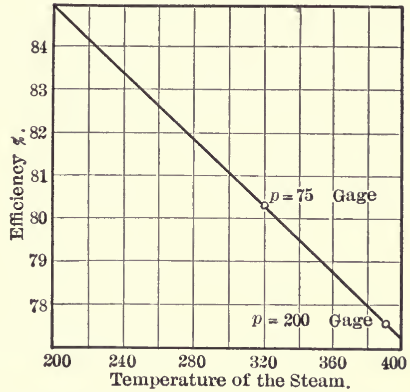


FIG. 137.—Relation between the efficiency and the temperature of evaporation.

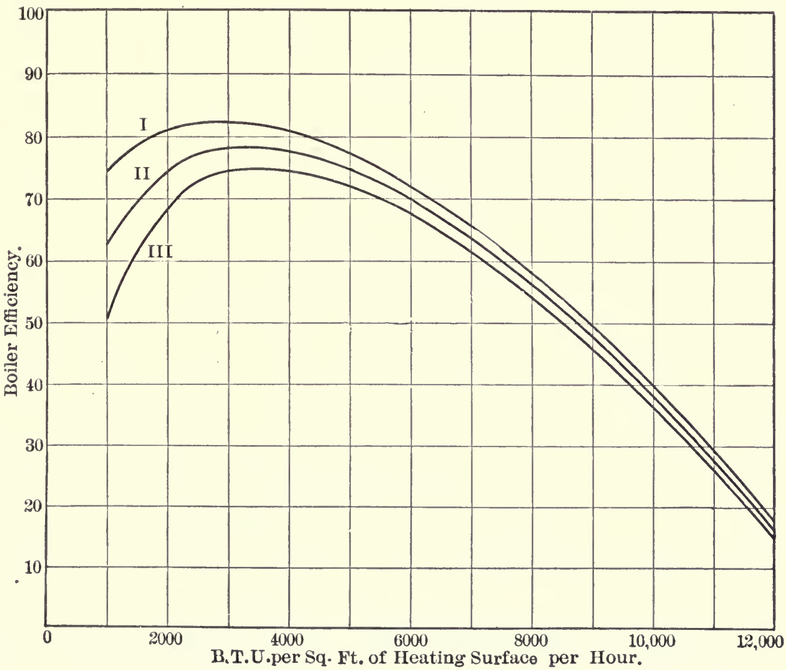


FIG. 138.—Relation between the efficiency and the rate of driving, allowing for radiation.

Curve I is for 5 per cent radiation loss.
Curve II is for 10 per cent radiation loss.
Curev III is for 15 per cent radiation loss.

to the two-thirds power of the heating surface. If we assume that this radiation loss in a boiler of normal design is 10 per cent of the heat generated when the boiler is operated at the normal rate of driving, we will have the relation between the efficiency and rate of driving shown in Fig. 138. Curves are added showing the relation of the efficiency and the rate of driving when the radiation loss is 5 per cent and also 15 per cent of the heat generated at normal load. It will be seen from these curves that there is a definite limit, depending upon the per cent of radiation loss, which determines the most efficient rate of driving and the proper allowance of heating surface per boiler horse-power. It will be seen that a boiler with an excess of heating surface may be practically less efficient than a smaller boiler which is operated at a higher rate of driving, besides being more costly. The rate of driving commonly adopted at the present time is that rate which experience shows to give the best efficiency.

251. Heat Losses in a Boiler Plant. The heat losses incurred in the operation of a boiler plant arise from four sources. The first source of loss is caused by incomplete combustion. Such loss is due to the dropping of unburned coal through the grates, the escape of unburned gases to the stack, etc. The second source of loss is the inefficiency of the heating surface. Loss from this source is usually termed **stack loss.** The third source of loss is radiation. The fourth source of loss is the latent heat of the water formed by combustion. Loss from this source is usually exceedingly small.

Improvement in the efficiency of the boiler plant must be looked for from the following sources: By improvement in the management of fires and the construction of furnaces we may increase the furnace temperature, reduce the quantity of air required per pound of fuel, and reduce the loss due to incomplete combustion. By careful arrangement and construction and properly clothing the boiler in non-conducting materials, we may reduce the radiation loss. Whenever the radiation loss is sufficiently reduced to warrant it, we may reduce the rate of driving by increasing the heating surface per boiler horse-power. Finally, we may so arrange the gas passages and heating surfaces that the gases are brought into thorough contact with the heating surfaces, thus increasing their conductivity.

252. Distribution of Losses as Shown by a Boiler Test. The following example will serve to show the distribution of losses in a boiler and the method of computing these losses from the results of a boiler test. The coal used in this test was shown by proximate analysis to contain 4.5 per cent of moisture, 16.0 per cent of volatile matter, 71.1 per cent of fixed carbon, and 8.3 per cent of ash. The heating value as obtained by the Parr calorimeter was 13,640 B.T.U. per pound. The analysis of the flue gas gave for CO_2 10.0 per cent, for O_2 9.8 per cent, for CO, 0.2 per cent;

and for nitrogen, by difference, 80.0 per cent. The total weight of water fed to the boiler was 2832 pounds. The total weight of coal fired was 469 pounds. 59.0 pounds of ash were taken from the ash pit at the end of the test. The average temperature of the feed-water was 73.5° F. The average steam pressure was 82.2 pounds absolute, and the average quality of the steam as shown by the throttling calorimeter was 98.7 per cent. The total heat available from the combustion of 469 pounds of coal was 6,390,000 B.T.U. From the analysis of this coal 8.3 per cent or 39 pounds is incombustible. Since 59 pounds of ash fell through the grates during the test, 20 pounds of this must have been unburned carbon. The potential heat contained in this 20 pounds of unburned carbon is 290,000 B.T.U. The heat transferred to each pound of water evaporated from 73.5° and at 82.2 pounds absolute will be, since the quality is 98.7 per cent, $898.8 \times .987 + 284 - 41.55 = 1120$ B.T.U. The heat transferred to the entire quantity of water evaporated is $2832 \times 1120 =$ 3,170,000 B.T.U.

The remainder of the heat then passed up the chimney in potential form in unburned gases, or was carried away in the sensible and latent heat of the flue gas, or was radiated into the boiler room and so lost.

From the flue gas analysis, we find that the number of c.c. of oxygen accounted for in 100 c.c. of flue gas will be

$$9.8 + 10 + \frac{0.2}{2} = 19.9 \text{ c.c.}$$

The oxygen represented by the nitrogen is

$$.261 \times 80 = 20.87 \text{ c.c.}$$

The difference, or 1.0 c.c., united with hydrogen to form water. The number of pounds of carbon burned per pound of oxygen supplied was equal to

$$1.44\left(\frac{10.0 + 0.2}{80}\right) = .2014.$$

The number of pounds of hydrogen burned per pound of oxygen supplied was

$$\frac{.48 \times 1.0}{80} = .0060.$$

The excess of air was

$$\frac{383 \times 9.8}{80 - 3.83 \times 9.8} = 81.6\%.$$

The number of pounds of air supplied per pound of combustible was

$$\frac{4.35}{.2074} = 21.$$

The number of pounds of nitrogen in the flue gas per pound of combustible was

$$\frac{3.35}{.2074} = 16.17.$$

The number of pounds of carbon dioxide in the flue gas per pound of combustible was

$$\frac{5.27 \times 10.0}{80 \times .2074} = 3.18.$$

The number of pounds of carbon monoxide was

$$\frac{3.36 \times 0.2}{80 \times .2074} = 0.04.$$

The number of pounds of water vapor was

$$\frac{4.20 \times 1.0}{80 \times .2074} = .25.$$

The number of pounds of free oxygen was

$$\frac{3.83 \times 9.8}{80 \times .2074} = 2.27.$$

Adding the water equivalents of these various gases we will have 5.274. The latent heat of evaporation of the water vapor will be 262 B.T.U.

Each pound of combustible shown by the flue gas consists of

$$\frac{.2014}{.2074} = .971 \text{ lbs. of carbon,}$$

and

$$\frac{.0060}{.2074} = .029 \text{ lbs. of hydrogen.}$$

The heating value per pound of combustible will then be

$$.971 \times 14500 + .029 \times 62000 = 15880 \text{ B.T.U.}$$

Of the coal burned in the furnace 20 pounds, or 4.26 per cent, dropped through the grates in the form of unburned carbon. The amount of heat lost in this manner was $.0426 \times 14500 = 620$ B.T.U. per pound of coal, leaving $13640 - 620 = 13020$ B.T.U. per pound of coal due to the burning of combustible substances appearing in the flue gas. Dividing this quantity by 15,880 we will have .820 pounds of combustible accounted for in the flue gas for 1 pound of coal fired. The total weight of combustible accounted for in the flue gas will therefore be $.820 \times 469 = 384.5$ pounds.

Since the boiler room temperature is 70°, and the stack temperature 765°, the flue gases will be rejected at a temperature 695° higher than their original temperature. The sensible heat carried away in the flue gases may be found by multiplying this difference in temperature by the water equivalent of the flue gas per pound of combustible and the product by the number of pounds of combustible shown by the test to be present in the flue gases. This gives

$$695 \times 5.274 \times 384.5 = 1{,}410{,}000 \text{ B.T.U.}$$

The latent heat carried away by the water vapor in the flue gas will be

$$262 \times 384.5 = 101{,}000 \text{ B.T.U.}$$

The number of B.T.U. lost as potential heat in the CO in the flue gas will be $.04 \times 384.5 \times 4380 = 67{,}400$ B.T.U. Adding together the heat lost in the flue gas, the heat imparted to the water and the heat lost by incomplete combustion, we will

have 5,038,000 B.T.U. Subtracting this from 6,390,000 B.T.U. which was the total heat supplied, we will have the radiation loss, which was 1,352,000 B.T.U. Expressing these various heat losses as percentages of the total heat in the coal fired, we will have 4.54 per cent, for the loss through the grates, 49.6 per cent of the total heat imparted to the water, 22.05 per cent for the loss in the sensible heat of the flue gas, 1.58 per cent for the loss in the latent heat of the water vapor in the flue gas, 1.05 per cent for the loss in the unburned CO and 21.2 per cent for the radiation loss. The radiation loss in this case was unusually high, since the boiler was a vertical fire tube boiler and was not protected by any non-conducting covering, the plates of the boiler being exposed to the air of the fire room. The distribution of heat may be tabulated as follows.

	B.T.U.	Per Cent.
Heat supplied	6,390,000	100
Heat utilized	3,170,000	49.6
Stack loss: Sensible heat	1,410,000	22.05
Stack loss: Latent heat	101,000	1.58
Incomplete combustion; coal through grates	290,000	4.54
Incomplete combustion; loss in CO	67,400	1.05
Radiation and error	1,352,000	21.18

The efficiency of a grate is found by subtracting from 100 per cent the heat loss in per cent due to the unburned carbon which drops through the grate. In this case the efficiency of the grate was $100-4.54=95.46$ per cent. The efficiency of the boiler and grate is found by dividing the heat imparted to the water evaporated by the total heat in the coal fired. The efficiency of the boiler and grate in this case was 49.6 per cent. The efficiency of the boiler is found by dividing the efficiency of the boiler and grate by the efficiency of the grate. In this case the efficiency of the boiler was 51.9 per cent.

PROBLEMS

1. A boiler evaporates steam at a temperature of 400° F. 20 lbs. of air are used per pound of coal (i.e., 1 lb. of coal produces 21 lbs. of flue gas). The temperature of the furnace is 2400° F. Assuming that the value of B in Eq. (6) in Art. 242 is 5.5 and that coal is burned at the rate of 1 lb. for every 4 sq.ft. of heating surface, find the probable final temperature of the flue gas. Ans. 578° F.

2. What is the theoretical efficiency of the heating surface in the above problem? Ans. 78%

3. A boiler contains an aggregate of 1400 sq.ft. of heating surface. What is its nominal horse power? Ans. 117.

4. How many lbs. of water will it evaporate per hour into steam of 98 per cent quality at a pressure of 100 lbs. gage from feed-water at a temperature of 70°, at rated load. Ans. 3,460 lbs.

5. How many square feet of grate surface will be required for the above boiler, if 20 lbs. of coal of 13,000 B.T.U. are burned per square foot of grate, and the boiler is assumed to be of 70 per cent efficiency. Ans. 21.5 sq. ft.

6. What will be the final temperature of the flue gases in Problem 1, if the rate of combustion be doubled? Ans 997° F.

7. What will be the theoretical efficiency of the heating surfaces in this case? Ans. 64.5%.

CHAPTER XVI

BOILER PLANT AUXILIARIES

253. The Chimney. Height Required. The chimney is a device for producing a **draft** or difference of air pressure, which is utilized to force air through the fire, and thence through the furnace, the boiler itself, and the breeching through which the furnace gases are discharged into the chimney. The chimney depends for its operation upon the difference in weight of the column of gas which it contains, and of a column of equal height and cross-section of the outside air. The weight of a column of flue gas or air of unit cross-section is proportional to the barometric pressure, to the absolute temperature of the gas or air, and to the height of the column. Let H be the height in feet of the top of the chimney measured from the grates, T_a be the absolute temperature of the atmosphere, T_c be the mean absolute temperature of the chimney gases, B be the normal barometric reading in inches of mercury for the region in which the chimney is erected, N be the per cent of CO_2 in the flue gas, and D be the draft produced or required, measured in inches of water. The weight of one cubic foot of air will be, from the characteristic equation of gases

$$W = \frac{P}{53.34\,T}, \quad \quad (1)$$

the pressure in pounds per square foot will be

$$P = 70.721B. \quad \quad (2)$$

The density of carbon dioxide is 1.52× that of air. Consequently, the density of the flue gas will be increased by .0052 for every per cent of carbon dioxide present. Therefore the weight of one cubic foot of flue gas will be

$$W = \frac{70.7\,B}{53.3\,T_c} \times (1 + .0052\,N) = \frac{1.329\,B\,(1 + .0052\,N)}{T_c}\text{ lbs.,} \quad \quad (3)$$

and the weight of a column one foot square and H feet high will be

$$W = \frac{1.329\,B\,H\,(1 + .0052\,N)}{T_c}\text{ lbs.} \quad \quad (4)$$

The weight of a column of external air, one square foot in cross-section, and H feet high, will be

$$\frac{1.329\,B\,H}{T_a}\text{ lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (5)$$

The difference in pressure produced at the base of the chimney in pounds per square foot is therefore

$$1.329\,B\,H\left(\frac{1}{T_a}-\frac{1+.0052\,N}{T_c}\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (6)$$

Since a column of water one inch high produces a pressure of 5.19 pounds per square foot, the draft produced by the chimney will be

$$D = .256\,B\,H\left(\frac{1}{T_a}-\frac{1+.0052\,N}{T_c}\right). \quad . \quad . \quad . \quad . \quad . \quad . \qquad (7)$$

Solving the above equation for H, in order to find the height of chimney required to produced a given draft we will have

$$H = 3.905\frac{D}{B\left(\frac{1}{T_a}-\frac{1+.0052\,N}{T_c}\right)}. \quad . \quad . \quad . \quad . \quad . \quad . \qquad (8)$$

In the absence of more definite data we may assume that in most actual cases of chimneys serving boiler plants without economizers, under normal atmospheric conditions we will have for the draft produced.

$$D = .007\,H, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (9)$$

and for the required height of chimney,

$$H = 140\,D. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (10)$$

254. Draft Required by a Boiler Plant at Nominal Load. When air or gas flows through a restricted passage, a difference of pressure is required to give it motion, both to give the air velocity, and in order to overcome friction. It follows, therefore, that in its passage through the fire, the boiler, the damper, the breeching and the chimney, the flue gas encounters at every point a resistance to its motion. The resistance is measured by the difference in pressure required to move the gas through the passage considered, and is shown by experiment to be nearly proportional to the 1.8 power of the quantity of the gas passing in a given time. When a boiler and furnace of ordinary design are operated at their rated capacity, we find that the difference in pressure required to draw air through the fire is from .10 to .30 inches of water. The former figure is for free-burning bituminous coal consumed at the rate of 24 pounds of coal per square foot of grate, and the latter figure is for anthracite buckwheat consumed at half this rate. These are the

usual rates of combustion in furnaces properly designed for burning these fuels when operated at their rated capacity. The resistance of the furnace and boiler passages to the current of gases, ranges from .15 to .30 inches of water, approaching the lower value in the case of small boilers and of water tube boilers, and the higher one in the case of large boilers and of fire tube boilers. Unless the breeching is exceptionally long and tortuous, its resistance will range from .05 to .20 inches of water. The resistance offered by the chimney itself, when properly designed and operating at rated load, ranges from 10 per cent to 20 per cent of the total draft produced, and includes the difference in pressure required to produce the actual velocity of the gases in the chimney. The total draft required of a chimney operating a boiler plant at rated load ranges from .40 to 1.00 inches of water, according to the character of the plant and the kind of fuel, and in most practical cases the draft required will be between .55 and .75 inches of water.

255. Draft Required by a Boiler Plant when Operating at an Overload. If, however, it is desirable or necessary to operate the plant, or any part of it, at more than rated load, the chimney must be of sufficient height to provide a greater draft than is called for in normal service. After the draft required to operate the plant at rated load has been ascertained, we may compute the maximum draft required by the formula

$$D = D_r \left(\frac{\text{Maximum Load}}{\text{Rated Load}} \right)^{1.8}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

in which D is the draft required for the maximum overload, and D_r is the draft required at rated load. From the above it will be seen that at 25 per cent overload the draft required will be $1.50D_r$, at 50 per cent overload it will be $2.08D_r$, and at 100 per cent overload it will be $3.50D_r$. In practice it has been found that the minimum height of chimney which will give satisfaction with plants of normal design, ranges from 80 feet in the case of free-burning bituminous coal to 200 feet in the case of anthracite slack; heights which give under ordinary atmospheric conditions a draft of from .55 to 1.4 inches of water and permit overloads of about 10 per cent. It is usual in the case of plants burning anthracite coal of small size to assist the chimney by a steam jet or fan blower. The jet or blower furnishes the excess of pressure necessary to force the air through the fire, while the chimney serves only to draw the gases through the boiler passages and breeching and prevent their escape into the fireroom. Were it not for this, the height of chimney required to produce a reasonable overload capacity in such plants would be excessive. Chimneys above 200 feet in height are not therefore usually required in practice, nor are they economical to build.

They are sometimes necessary, however, in order to discharge smoke or noxious gases at such a height that their discharge will be harmless.

It may be remarked that when a boiler plant operates at an overload, the temperature of the chimney gases is increased, which makes the overload capacity of a chimney rather more than theory would indicate. If the top of a chimney is properly formed, wind increases the draft, and if there is no wind, the column of hot air rising above the chimney acts to increase its effective height. Chimneys, especially hort chimneys of large cross-section, may therefore be expected to give a somewhat greater draft than theory would indicate.

256. Required Diameter of Chimneys. From experiment it is known that the loss of pressure which a fluid suffers in passing from one point to another in a tube of uniform diameter is directly proportional to the distance between the points and to the 1.8 power of the velocity of the fluid, and inversely proportional to the 1.3 power of the diameter of the tube. This loss also depends upon the density and viscosity of the fluid and the character of the inner surface of the tube, but the proportionality factor for any given tube and for any given fluid of a constant density is a constant quantity. Applying this law to the case of a chimney we will find that the draft required to produce a given velocity of the chimney gases, will be

$$D_c = K_1 H \left(\frac{V^{1.8}}{d^{1.3}}\right), \quad \ldots \ldots \quad (1)$$

in which D_c is the draft, in inches of water, required to produce the given velocity; H is the height of the chimney in feet, V is the velocity of the chimney gases in feet per second, and d is the diameter of chimney in inches. Solving equation (1) for V we will have

$$V = \sqrt[1.8]{\frac{D_c\, d^{1.3}}{K_1\, H}}. \quad \ldots \ldots \quad (2)$$

If we make the allowable loss of draft in the chimney some fraction of the total draft which the chimney will produce, then since the total draft is proportional to the height of the chimney, we may replace $\frac{D_c}{H}$ by a constant, and obtain the equation

$$V = K_2 \sqrt[1.8]{d^{1.3}} = K_2\, d^{.72}. \quad \ldots \ldots \quad (3)$$

Since the capacity of the chimney is proportional to the product of the velocity of the flue gases and the area of the cross-section (which is proportional to d^2) we may write

$$HP = K_3(d^{2.72}). \quad \ldots \ldots \quad (4)$$

In which HP is the nominal capacity of the chimney, in terms of the boiler horse-power which it will serve. Solving equation (4) for the diameter of the chimney, we will have

$$d = K\,(HP)^{.37}. \qquad (5)$$

A study of chimneys of good proportion giving satisfactory service shows that the value of the constant K should be about 5.5, in which case the resistance of the chimney to the passage of the gases, at rated load, is about 15 per cent of the total draft produced, in the case of plants of rather poor economy. Therefore in order to find the total draft required in the case of a chimney designed by this formula, we must multiply the draft found by equation (1) in Art. 255 by 1.17. In order to facilitate computation equation (5) may be written in the form

$$\log d = .37 \log HP + .75. \qquad (6)$$

in which d is the diameter of the chimney in inches, and HP is the nominal horse-power of the boiler plant which it is to serve.

257. Effect of Overloading the Boiler Plant upon the Capacity of the Chimney. In case a chimney designed by the above rule is called upon to handle a greater quantity of flue gas than that for which it was designed, more than 15 per cent of the total draft must be utilized in overcoming the resistance of the chimney itself. Since the resistance of the chimney is proportional to the 1.8 power of the velocity of the gases, the amount of draft required to overcome the resistance of the chimney will be

$$D_c = .15\,D\left(\frac{\text{actual load}}{\text{rated load}}\right)^{1.8},$$

in which D_c is the draft required to overcome the resistance of the chimney and D is the total draft produced by the chimney.

It has already been noted that chimneys are usually made of sufficient height to provide a small overload capacity, usually from 10 to 25 per cent. If it is desired to increase the power of a boiler plant and yet to use the old chimney in carrying away the gases, this excess of height affords an opportunity for increasing the capacity of the plant by reducing slightly the overload capacity of the individual units. If D equals the total draft available, in inches of water, and D_f be the amount of draft required to overcome the resistance of the fire, the boiler and the breeching when the individual boilers are operated at the desired rating, the difference is available for overcoming the resistance of the chimney. If this difference is greater than 15 per cent of the total draft produced, the capacity of the chimney will exceed that given by Equation (4), Art. 256., and the ratio of the capacities will equal the 1.8 root of the ratio of the draft actually available for overcoming the chimney resistance to that assumed in equation (6) to be available. Writing this as an equation we will have

$$HP_a = HP_r \sqrt[1.8]{\frac{D - D_f}{.175\,D_f}},$$

in which HP_a is the nominal rating of the boilers which the chimney will serve under the assumed conditions of overload, HP_r is the rating of the chimney from formula

4 Art. 256, D is the total draft the chimney will produce, and D_f is the resistance of the furnace, boiler and breeching, when operated under the assumed conditions of overload.

258. Example of Chimney Design. The following example will serve to make clear the application of the principles developed above. A chimney is required to burn buckwheat anthracite in a region where the normal barometer reading is 28 inches, and the normal summer temperature 70°. The plant will be required to develop about 25 per cent overload and is of 1000 horse-power nominal capacity. Assume the temperature of the gases to be 500° F., and the per cent of CO_2 to be 10 per cent.

The draft required by the fire at normal load $= .30''$.

The draft required to overcome the boiler resistance at normal load $= .25''$.

The draft required to overcome the resistance of the breeching at normal load $= .10$.

The sum of the above $= .65''$.

Draft required to overcome resistance of chimney $= .65'' \times .175 = .12''$.

Total draft required at rated load $= .77''$.

Total draft required at 25 per cent overload $= .77'' \times 1.25^{1.8} = 1.15''$.

Substituting the proper values in formula 8, Art. 253, for the height of chimney, we will have

$$H = 3.905 \frac{1.15}{28\left(\frac{1}{530} - \frac{1 + .0052 \times 10}{960}\right)} = 194 \text{ ft.}$$

In order to find the diameter of this chimney, we use equation (5), Art. 256, which gives

$$d = 5.5 \times 1000^{.37} = 72 \text{ inches.}$$

Should it be desirable to extend this plant at any future time, its power may be greatly increased, provided the individual units are not required to carry 25 per cent overload. If they are required to carry rated load merely, the nominal power of the station will be, according to the formula in Art. 257,

$$HP = 1000 \sqrt[1.8]{\frac{1.15 - .65}{.175 \times .65}} = 2280 \text{ horse-power.}$$

259. The Injector. An injector is an apparatus for forcing water against pressure by utilizing the impact of a jet of steam. The apparatus is shown in principle in Fig. 139, in which A is a pipe supplying steam to the nozzle B, in which the steam, expanding adiabatically, acquires a high velocity. The jet of steam from this nozzle passes through the tube C, which is known as a combining tube, and in doing so carries with it any air in the neighborhood. As a result a vacuum is created which sucks the water through the suction pipe D into the chamber surrounding the jet. As soon as the steam comes in contact with the water so drawn in, it is instantly condensed and imparts its momentum to the water, forcing it into the combining tube. The water issuing from the combining tube in the form of a jet passes into a third tube E, which is known as the discharge tube, in which its velocity is trans-

formed into pressure. After passing through the discharge tube, the water flows through a check valve and into the boiler or other region into which it is desired to force it. Before the current of water is established the water escapes around the space between the combining and the discharge tubes and flows away through an opening F, termed the overflow. As soon, however, as a solid stream of water of sufficiently high velocity is flowing through the combining tube, no water will escape through the overflow.

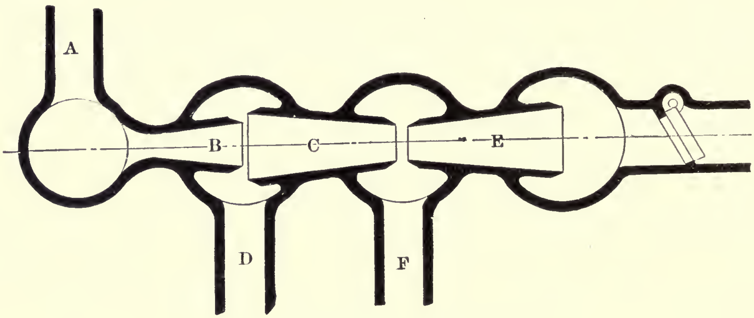


FIG. 139.—Diagram of an injector.

260. Efficiency of the Injector. In order to illustrate the principle of the injector, we will assume one taking dry and saturated steam at a pressure of 100 pounds absolute and delivering the water against the same pressure. Assume that the absolute pressure within the suction chamber is 10 pounds per square inch. The kinetic energy of the jet, per pound of steam flowing, is 127,900 foot-pounds, its velocity is 2840 feet per second, and its momentum 2840 pounds feet per second. The head against which the water is discharged is $(100-10)\times 2.31 = 208$ feet. Allowing 25 per cent excess for friction we will have, say, 250 feet. The velocity of the jet issuing from the combining tube must be

$$V = \sqrt{2\,g\,h} = \sqrt{64.4 \times 250} = 128 \text{ ft. per sec.}$$

The momentum of the jet, which is composed of x pounds of water and 1 pound of steam, must from the principles of mechanics, be equal to that of the steam jet. Hence

$$(1+x)128 = 2840,$$

and $x = 21.2$ pounds of water pumped per pound of steam supplied.

The work done per pound of steam is

$$21.2 \times 250 = 5300 \text{ ft.-lbs.}$$

The efficiency of the injector as a pump, is therefore only

$$\frac{5300}{127400} = 4.15 \text{ per cent,}$$

of the efficiency of a Rankine cycle engine working through the same pressure range. It will be seen from the above example that as a pump the injector is very inefficient, although all of the heat of the steam is returned to the boiler. Of the kinetic energy of the steam jet, only from 3 to 6 per cent is available to force the water into the boiler, the remainder being transformed into heat as a result of the inelastic impact of the steam upon the water.

The injector is often a troublesome instrument to operate, since the condition of the apparatus must be perfect before it will give satisfactory service. The stoppage of any of its small passages by a piece of coal or by the accumulation of scale

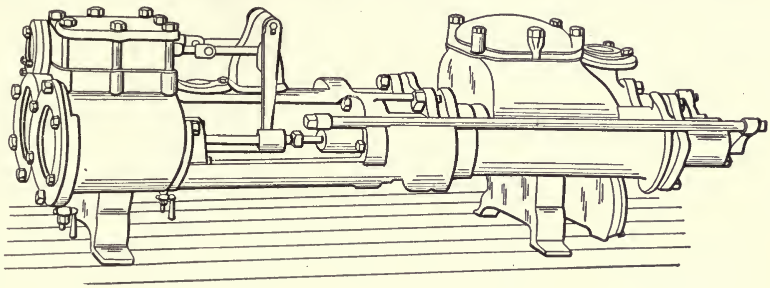


FIG. 140.—Boiler feed pump.

renders it inoperative. It is commonly used in the case of locomotive and portable boilers, but is not usually used in stationary service. Many modifications of the injector are in use and are known by different names. In some modifications, one injector is arranged so as to deliver water to the suction chamber of a second injector, such an injector being known as an inspirator or a double tube injector.

261. Boiler Feed-pumps. Boilers are usually supplied with feed-water by means of a direct-acting steam pump, such as is illustrated in Fig. 140. These pumps are of various types, but almost all of them use steam non-expansively and are very wasteful. The exhaust from these pumps, however, is usually employed to heat the feed-water, and on

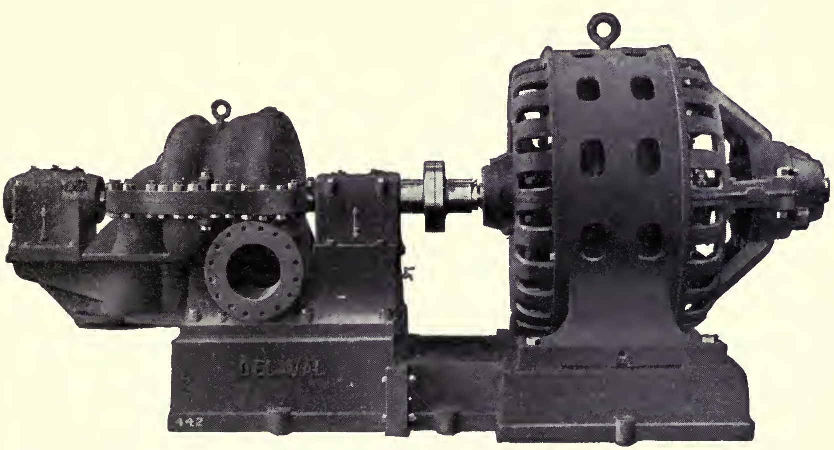


FIG. 141.

this account no loss is experienced from their use, since they return to the boiler all of the heat which is taken for their operation. Of late years, in places where economizers are installed, motor-driven centrifugal pumps have been used for boiler feeding. Such a pump is illustrated in Fig. 141. These pumps are, of course, practically as economical as

are the main engines themselves. They run at nearly constant speed, maintaining a constant pressure in the delivery pipe and the quantity of water delivered and power required automatically adjust themselves to the needs of the boiler for feed-water.

262. Feed-water Heaters. In order to avoid losses incident upon supplying a boiler with cold feed-water, the heat of the exhaust steam of the engines and pumps is usually utilized for heating the feed-water. A gain of from 10 to 12 per cent may be realized under usual conditions by using a feed-water heater. Feed-water heaters are of two kinds, known as closed and open heaters. In the closed heaters, the water to be heated is usually forced through tubes which are surrounded by the exhaust steam, which does the heating. Such a heater is shown in section in Fig. 142. Sometimes the water surrounds the tubes and the exhaust steam passes through them. In open heaters, the feed-water is brought into direct contact with the exhaust steam. Usually the water coming from an open heater is a few degrees hotter than that coming from a closed heater. The theory of heat transfer in the closed heater is the same as in the case of the surface condenser, and they may be designed by the same principles.

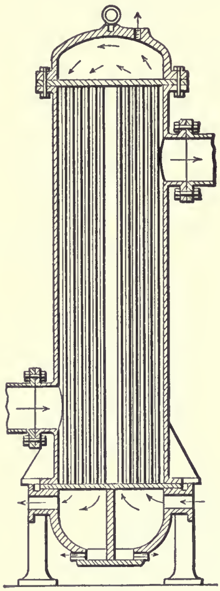


FIG. 142.—Section of a closed heater.

The following problem will serve to illustrate the economy obtained by the use of a heater. Assume that a boiler operates at a pressure 165 pounds absolute, that the feed-water supplied has a temperature of 70° and that by the use of a suitable heater the feed temperature may be raised to 205°. The total heat which must be imparted to the feed-water in evaporating it from 70° at 165 pounds is found by subtracting the heat of the liquid at 70° from the total heat of the steam at 165 pounds. In the same way, the heat required when the heater is used will be found by subtracting the heat of the liquid at 205° from the total heat of the steam. We find in one case 1156.9 B.T.U. are required, while in the other case only 1022.0 B.T.U. are required, showing that if the heater is adopted, the quantity of fuel required by the boiler in a given time will be reduced more than 11.7 per cent, since not only is less heat needed to evaporate the required quantity of water, but the rate of driving of the boiler is decreased, and therefore the efficiency of the boiler plant is improved.

263. The Economizer. The economizer is an apparatus which utilizes the waste heat of the flue gases to heat the water entering the boiler. It is possible by the use of a suitable economizer to reduce the temperature of the chimney gases to within 200° of the temperature of the feed-water and to heat the water entering the boiler practically to the temperature of vaporization. It will be seen that the use of the economizer will very greatly reduce the fuel required for a given quantity of steam generated. Economizers are usually built with cast-iron tubes and headers, and are provided with sliding rings which scrape the soot from the tubes. When an economizer is used, it is usually necessary to use a fan to create the necessary draft, since the temperature of the gases entering the chimney is not sufficiently high to cause a good draft.

The theory of the heat transfer in the economizer is different from that of the boiler, since the water entering the economizer is at a low temperature and it gradually increases as it passes through the economizer. The current of hot gases flows through the economizer in the opposite direction to the current of the water, so that the hottest gases come into contact with the water entering the boiler and the coolest gases into contact with the water entering the economizer. By this means the temperature of the gases leaving the economizer may be reduced below the temperature of the water entering the boiler.

The difference in temperature between the flue gases and the water varies at different points in the economizer, usually being greater at the end next the boiler than at the cool end of the economizer. The water equivalent of the flue gas discharged per second by the boiler is usually less than the weight of the feed-water supplied to the boiler in the same time. It will be seen therefore that the fall in temperature of the flue gases between any two points in the economizer will be greater than the rise in temperature of the feed-water and the less the excess of air used for combustion the more pronounced will be this effect.

It is usual to supply from $3\frac{1}{2}$ to 5 square feet of economizer surface per boiler horse-power. When an economizer is so proportioned, the feed-water entering the boilers is usually heated to about 300° F. The Greene Economizer Company give the following empirical formula for the rise in temperature of the feed-water in passing through an economizer:

$$x = \frac{y\,(T_1 - t_1)}{9.1 + \frac{5\,w + G\,C}{2\,G\,C}\,y},$$

in which x = the rise in temperature of the feed-water;

T_1 = the temperature of flue gas entering the economizer;

t_1 = temperature of feed-water entering economizer;

w = pounds of feed-water per boiler horse-power per hour;
G = pounds of flue gas per pound of combustible;
C = pounds of coal burned per boiler horse-power per hour;
y = square feet of economizer heating surface per boiler horse-power.

264. The Superheater. The use of superheated steam in connection with the steam turbine is becoming very common on account of the great gain in efficiency resulting therefrom. Superheaters may form a part of the boiler and be heated by the gases on their way through the boiler. Many superheaters, however, are independently fired, having their own furnace and gas passages. The use of a superheater in connection with a boiler plant does not directly affect the efficiency of the boiler plant. It does, however, permit the use of a small boiler plant, since less steam will be required from a boiler plant equipped with superheaters, on account of the greater efficiency of the engines. In case the size of the boiler plant is not reduced, the effect of the introduction of superheaters will be to reduce the rate of driving and so increase the efficiency of the boilers.

Superheaters are usually formed of heavy seamless steel tubing, expanded into forged steel headers and sometimes protected by cast-iron rings from the direct action of the hot gases. A superheater requires greater care in its operation than is usually required with a boiler, since the superheaterelements are not filled with water, and therefore are easily overheated. The conductivity of a unit area of superheater surface is considerably less than that of the same area of boiler-heating surface, since the heat is transferred from the metal to a gas or superheated vapor in the case of a superheater, while it is transferred from the metal to a liquid in the case of a boiler.

The amount of superheater surface required per boiler horse-power varies according to the required temperature of the steam, and the temperature of the gases in contact with the superheater surface. The following empirical formula has been proposed by J. E. Bell:

$$x = \frac{10\,S}{2\,(T-t)-S},$$

in which x = the number of square feet of superheater surface per boiler horse-power;
S = the superheat in degrees F;
T = the temperature of the flue gases at the point where the superheater is located;
t = the temperature of the saturated steam.

PROBLEMS

1. What height of chimney will be required to give a draft of 1.2 inches of water under ordinary operating conditions? Ans. 168 ft.

2. A boiler plant having a stack 100 ft. high will operate satisfactorily when evaporating 10,000 lbs. of water per hour. To what height must the stack be raised in order that the plant shall evaporate 12,000 lbs. of water per hour? Ans. 139 ft.

3. What diameter of chimney will be required for a boiler plant of 2000 horse-power? Ans. 91 inches.

4. A non-condensing engine is used in connection with a feed-water heater. Without the heater, the temperature of the feed is 65°. With the heater, the temperature of the feed is 190°. What per cent of coal is saved when the heater is employed? The boiler generates steam at a pressure of 80 lbs. gage and of 98% quality. Ans. 10.8%.

5. The feed-water heaters of a plant bring the temperature of the feed to 160°. When an economizer is employed the temperature of the feed is brought to 280°. What per cent of coal is saved by the economizer if the boilers produce dry and saturated steam at a pressure of 165 lbs. absolute? Ans. 11.2%.

6. An economizer is required to raise the temperature of the feed-water 100°. The temperature of the feed is 150°, the temperature of the flue gas entering the economizer is 600°, the boilers require 30 lbs. of feed-water per horse-power per hour, 4 lbs. of coal are burned per horse-power per hour, and 20 lbs. of flue gas are produced per lb. of coal. Find the number of square feet of economizer surface required per boiler horse-power. Ans. 2.98 sq.ft.

7. A superheater is required to superheat steam of a pressure of 180 lbs. absolute, 100°. The temperature of the gases is 1300°. Find the number of square feet of superheater surface required per boiler horse-power. Ans. 0.57 sq.ft.

CHAPTER XVII

WATER-COOLING APPARATUS

265. Advantage of Using Water-cooling Apparatus. Condensing water is usually taken from lakes or streams, or, at the seaboard, from the sea. When a power plant is erected at some point distant from a stream or other body of water, the plant may be made non-condensing, or it may be provided with some method for cooling the condensing water. A condensing plant not provided with any method for cooling water usually requires from 300 to 500 pounds of condensing water per horse-power per hour, and a non-condensing plant usually requires from 25 to 35 pounds of water per horse-power per hour, while a condensing plant equipped with a cooling apparatus will require only from 12 to 18 pounds of water per horse-power per hour. Hence, in all those cases where the plant is large and cooling water is expensive, either because only small quantities of water are available, or city water must be purchased, water-cooling apparatus is a necessary adjunct to the plant. The use of the cooling tower or other water-cooling device not only permits of economy in the consumption of water, but also of fuel, since a condensing plant will use only from one-half to two-thirds of the quantity of fuel required by a non-condensing plant of the same power.

266. The Cooling Pond. The simplest method of cooling condensing water is to construct an artificial pond having an impervious bottom from which the cooling water is drawn, and into which it is discharged after passing through the condenser. The evaporation from the surface of the pond and radiation of heat from the water into the surrounding air will keep the temperature of the water down in spite of the fact that heat is being continually added to it. The water must of course be replaced as fast as it evaporates. The pond must be of sufficient area so that the evaporation may go on at a rate which will dispose of the heat imparted to it without allowing the temperature of the cooling tower to become too great. The area required will depend upon the mean summer temperature, upon the dryness of the air, upon the character and amount of the load of the station, upon the depth of the water, and upon the amount of wind which may ordinarily be expected in the region. The rate of evaporation and consequently the rate at which the water is cooled is, of course, proportional to the area of water surface exposed, so that,

other things being equal, the area of the pond must be proportional to the power of the station. The higher the summer temperature of the region, the warmer the water in the pond must be for a given rate of evaporation per square foot of surface. During cold winter weather, although evaporation proceeds at a slower rate than in summer, the pond will lose heat more rapidly on account of radiation into the surrounding cold air. The humidity of the atmosphere and the amount of wind to be expected is also a very important factor in settling the size of the pond. If the air is dry, the water will evaporate rapidly from the surface of the pond, while if the climate is humid, the evaporation will proceed slowly. If the load upon the station is constant, the depth of water in the pond makes no particular difference. If, however, the load is variable, a much smaller pond may be made to serve if the water is deep, since this large mass of water will serve as a heat reservoir; absorbing the heat during the periods of peak load, with a small rise in temperature, and slowly recovering its normal temperature during the periods of light load.

267. Rate of Evaporation from the Surface of Water. The rate of evaporation per square foot of water surface exposed to air is, in theory, proportional to the difference between the quotient of the square root of the absolute temperature of the water into the density of saturated steam at that temperature, and the quotient of the square root of the absolute temperature of the air into the density of the water vapor present in the air.[1] We may, therefore, in engineering computations, take the rate of evaporation per square foot of surface as being proportional to the difference between the saturation pressure of water vapor at the temperature of the water and the actual pressure of the water vapor in the air. It has been found that in still air[2] the difference in pressure required to evaporate 1 pound of water per square foot of water surface per hour is about 3.2 pounds per square inch. In case a brisk wind is blowing over the surface, the rate of evaporation will be from four to six times as great as is given by the above rule. The following problem will serve to illustrate the principle:

Assume that the temperature of the water is 80°, that the temperature of the air is 70°, and that the humidity is 60 per cent. The saturation pressure at 80° is 0.505 pounds per square inch. The pressure of the water vapor in the atmosphere is $0.36 \times 0.60 = 0.22$ pounds per square inch. The difference in pressure is 0.285 pounds per square inch. Since the difference in pressure required to evaporate 1 pound of water per

[1] See Chapter XXVI.

[2] By still air is meant air in which the only currents are those produced by the difference in density of hot moist air and cool dry air. It does not mean air in which all currents are prevented by artificial enclosure.

square foot per hour is about 3.2 pounds per square inch, a difference of pressure of 1 pound will evaporate 0.30 pounds of water per square foot per hour in still air. The rate of evaporation from the surface will be in this case $0.285 \times 0.30 = 0.085$ pounds per square foot per hour in the case of still air.

268. Determining the Area of a Cooling Pond. It will be seen from the above that the area of a cooling pond required for a given station may be determined in the following manner. From the records of the Weather Bureau, the mean summer temperature and humidity for the region may be found: From these data, the mean pressure of the water vapor in the atmosphere may be determined from a steam table. Assuming the temperature to which the condensing water is to be cooled, find in the steam table the saturation pressure corresponding to this temperature and from it deduct the mean summer pressure of the water vapor in the air. Multiplying the difference so obtained by 0.30 we will have the quantity of water evaporated per hour per square foot of surface from the cooling pond. Dividing this quantity into the average quantity of steam rejected per hour by the engines of the station, we will have the required area of the cooling pond in square feet. Good condensing plants will on the average require an evaporation of 15 pounds of water per horse-power per hour from the cooling pond.

The total quantity of water which the pond must evaporate in twenty-four hours may be found by multiplying the twenty-four-hour factor of the station by its rated horse-power and their product by 15. In case it is known that the station is likely to be inefficient, it will be necessary to allow more than 15 pounds of water per hour.

The following problem will serve to illustrate the method of finding the area of a cooling pond. The mean summer temperature as obtained from the Weather Bureau reports is 75°, and the humidity 60 per cent. It is desired to cool the water in the pond to 80°. The station is to be of 1000 rated horse-power, and the twenty-four-hour load factor is 40 per cent. From the steam tables the pressure of saturated water vapor at 80° is 0.505, and at 75° temperature and 60 per cent humidity it is $0.429 \times 0.60 = 0.26$ pounds per square inch. For the difference we will have $0.505 - 0.26 = 0.245$. The rate of evaporation will then be $0.245 \times 0.30 = 0.073$ pounds per square foot. The average horse-power of the station will be $1000 \times 0.40 = 400$. The required rate of evaporation will be $400 \times 15 = 6000$ pounds per hour. The cooling surface will then be $\frac{6000}{.073} = 82{,}000$ square feet, or the pond required will be approximately 290 feet square.

The cooling surface allowed in the above pond is 82 square feet per rated horse-power. Since 82 square feet of water will weigh about 5100

pounds for every foot in depth, it will be seen that a comparatively shallow pond will serve to carry great overloads for a short period of time. Allowing 600 pounds of condensing water per horse-power per hour, it will be seen that the above pond will contain $8\frac{1}{2}$ hours' supply of condensing water for every foot of depth. It is generally well to make the pond deep enough to carry from twelve to twenty-four hours' supply of condensing water, allowing 600 pounds per horse-power per hour at the rated horse-power of the station. If the above pond be made 2 feet deep, the quantity of water contained will be ample.

It may be seen from the above computations that a cooling pond must be quite large. For small powers, say up to 500 horse-power, the cooling pond is a cheap and simple method of solving the problem of condensing water supply, provided land is cheap. In case land is expensive or the station is large, the spray pond or cooling tower will be a preferable method for obtaining a supply of condensing water.

269. The Spray Pond. A second method of cooling condensing water is by spraying the water into the air over the surface of the small pond, which may be, and quite often is, placed upon a roof of a building. In case the water is sprayed into the air in this way, the required area of the pond is very much less than when spraying is not used, but since the quantity of water contained in the pond is small, it is necessary to provide sufficient capacity to take care of the peak load of the station.

270. Area of Spray Pond Required. When water is sprayed in this manner, it will be cooled by evaporation from the surface of the drops, and since the surface exposed by the drops is vastly greater than the surface of the same quantity of water exposed in a pond, the evaporation will be very much more rapid. The final temperature of the water will, of course, be higher than the temperature of the dew point, and it will be found, as a usual thing, that the pressure of saturated water vapor of the final temperature of the water will be about 0.15 pounds per square inch higher than the pressure of the water vapor in the atmosphere. Thus, if water be sprayed into air having a temperature of 70°, and a humidity of 70 per cent, the pressure of the water vapor in the air will be $0.70 \times 0.36 = 0.25$ pounds per square inch. The temperature of the water will then be reduced to the temperature corresponding to the pressure of $0.25 + 0.15 = 0.40$ pounds per square inch. From the steam tables, this temperature is 73°. In case the drop in temperature of the water is large, (i.e., above 40 or 50° F.), it will be found that air in the neighborhood will become so saturated with moisture that the evaporation will not take place freely, in which case it may be necessary to spray the water twice in order to bring it to a sufficiently low temperature. It will usually be found preferable to use such a quantity of water that the required reduction in temperature will not exceed 40°. One square foot of pond

surface will be sufficient for the cooling of 200 pounds of water (about 3 per cent of which will be evaporated) and the spray nozzles should be placed a sufficient distance apart so that each will be allowed the area given by the above rule. It will usually be found that 3 square feet of pond surface per horse-power will suffice, but the total surface provided on such a basis must be estimated on the maximum and not the rated or mean horse-power of the station. If the efficiency of the station is low, so that the quantity of heat rejected per horse-power is more than is required to evaporate 15 pounds of water, the surface allowed per horse-power must be suitably increased.

271. Power Required by Spray Nozzles. When condensing water is cooled by the use of a spray pond, it is necessary to pump the water to the spray nozzles at a pressure of from 15 to 20 pounds per square inch. This takes a considerable amount of power. Assuming that 3 per cent is evaporated and that the quantity is 15 pounds per horse-power per hour, it will be seen that the quantity pumped per horse-power will be about 500 pounds. If this water be pumped against a head of 46 feet (which corresponds to a pressure of 20 pounds per square inch) and the efficiency of the pumping plant be 60 per cent, the work required to do this pumping will be 38,300 foot-pounds per hour, or 0.019 horse-power. The power required for pumping will therefore be between $1\frac{1}{2}$ and 2 per cent of the power of the station. In case steam pumps are used for this purpose, and they are run condensing, an extra allowance of water must be made for them, since such pumps are much less efficient than large engines, and the estimated evaporation of 15 pounds of water per horse-power per hour will not be sufficient to furnish them with cool condensing water.

272. The Cooling Tower. In large plants the cooling tower is the preferred method of providing a supply of condensing water. Cooling towers are divided into two classes, known as natural draft and mechanical draft towers, according as to whether the air is drawn through the tower by a chimney-like action, or forced into the tower by means of a fan or other form of mechanical impeller. In theory, the action of the cooling tower is as follows: the water coming from the condensers, which usually has a temperature of approximately 100°, is introduced at the top of the tower, which is filled either with wooden or tile checker work or heavy galvanized iron wire partitions. As the water descends through the tower, flowing over the checker work or wire mesh, it exposes a large area to the action of the air which is flowing upward through the tower. The air entering the tower has the temperature and humidity of the outdoor air. As it ascends through the tower, coming into contact with warm water, it chills this water by the evaporation of a small portion of it, and finally leaves the top of the tower at almost the temperature of the

entering water and laden almost to the saturation point with moisture. Since it is warmed as it ascends through the tower, it expands in volume and consequently is able to hold more moisture than it would were it not for this expansion. The addition of the water vapor which it absorbs also increases the volume. A portion of the heat taken from the water is carried away in the form of sensible heat in the air, on account of its rise in temperature. The most of it, however, is carried away in the form of latent heat, on account of the evaporation of a portion of the water. The following problem will serve to make clear the action of such a cooling tower.

273. Capacity of a Cooling Tower. Assume that a cubic foot of air enters the tower at a temperature of 70° and a humidity of 70 per cent. Were this air saturated with moisture, we find from the steam tables that the pressure of the water vapor present would be 0.36 pounds per square inch. The actual pressure of the water vapor will be 70 per cent of this or 0.25 pounds per square inch. The pressure of the dry air is therefore $14.70-0.25=14.45$ pounds per square inch. The quantity of moisture contained in the air is $0.001148\times0.70=0.000804$ pounds per cubic foot. From the equation $PV=X\,R\,T$ we find the weight of one cubic foot of dry air to be

$$\frac{14.45\times144}{53.3\times530}=0.0737 \text{ lbs.}$$

The total heat of the moisture contained in the air is found by adding the heat of superheat to the total heat at the temperature corresponding to the pressure of the water vapor. Since the superheat is 11°, the specific heat of superheated steam of this temperature is 0.46, the total heat is 1085.4 B.T.U. per pound and the weight of moisture in the cubic foot of air is 0.000804, we will have for the total heat of the moisture in 1 cubic foot of air, the value

$$.000804\ (11\times0.46+1085.4)=0.876 \text{ B.T.U.}$$

Let us assume further that the air comes from the cooling tower at a temperature of 100° and saturated with moisture. The pressure of the moisture contained in the air is now 0.946 pounds per square inch, which leaves as the pressure of the dry air 13.753 lbs. per square inch. The volume of what was 1 cubic foot of air will now be increased, the new volume being to the old volume inversely as the absolute pressure of the dry air and directly as its absolute temperature. We therefore have for the new volume of this quantity of air

$$\frac{14.45}{13.75}\times\frac{560}{530}=1.11 \text{ cu.ft.}$$

This quantity of air will contain $0.002851 \times 1.11 = 0.00316$ pounds of water-vapor at a temperature of 100° when saturated, and the total heat of this vapor will be $1103.6 \times 0.00316 = 3.49$ B.T.U. It will be seen that the amount of heat carried away in the water vapor is equal to $3.49 - 0.88 = 3.61$ B.T.U. for each cubic foot of air introduced in the tower. The air itself was heated from a temperature of 70° to a temperature of 100°, and this 30° rise in temperature added to it $0.0737 \times 30 \times 0.238 = 0.525$ B.T.U., a quantity found by multiplying the weight of the air by its rise in temperature and the product by the specific heat of the air at constant pressure. The total quantity of heat carried away by each cubic foot of the air introduced into the tower is then $0.520 + 2.61 = 3.14$ B.T.U.

When a cooling tower is working at approximately its rated capacity the water will be cooled until its temperature is about that of saturated water vapor having a pressure from 0.15 to 0.25 pounds per square inch higher than that of the water vapor in the entering air. The air will leave the tower with a temperature from five to ten degrees lower than the entering water and with a humidity of from 90 to 100 per cent. We are usually safe therefore in assuming that each cubic foot of air delivered to the tower will carry away at least 2.5 B.T.U. from the condensing water. Since a condensing steam plant of good economy will reject from 10,000 to 15,000 B.T.U. per horse-power per hour, it will be seen that we must supply to the cooling tower from 60 to 100 feet of air per minute for each horse-power developed by the plant.

274. Method of Designing a Cooling Tower. In designing a cooling tower it is necessary to provide a sufficient surface of checkerwork to evaporate the required quantity of water; to provide a sufficient cross-section in the air passages so that the pressure required to circulate the air through the tower will not be excessive; to arrange the water distribution system so that the water will be distributed evenly; to arrange the air passages so that the supply of air will be distributed uniformly to all parts of the checkerwork; and to provide means for moving the air and pumping the water. In case the distribution of water or air is uneven, the efficiency and capacity of the tower will be seriously impaired.

The first point to be determined in cooling tower design is the area of checker work which must be exposed to the action of the air. The rate of evaporation from the surface of checker work in a cooling tower having forced draft is about five times that which occurs in the case of an open pond exposed to still air. The rate of evaporation per square foot of surface per hour will therefore be found by multiplying the difference between saturated water vapor of the temperature of the water leaving the tower and the actual pressure of the water vapor in the air entering the tower by 1.5. Having found the rate of evaporation and knowing the temperature of the water entering the tower, we may compute by the

converse process, the approximate temperature and humidity of the air leaving the tower, and from this we may determine the quantity of heat and of moisture carried off per cubic foot of air supplied. From the total heat rejected by the engine when working at its rated load, we may determine the total quantity of air required, and the total area of the checker work. The depth of the checker work will depend on the available draft in case the tower is a natural draft tower and may be given any reasonable value in the case of a forced draft tower. It is usual to make the depth of checker work in the latter case twice the least dimensions of the base of the tower.

275. Example of the Design of a Cooling Tower. We will assume the following problem to illustrate the method of cooling tower design. Temperature of the air entering 75°, humidity 70 per cent, required temperature of condensing water 80°, temperature of water entering the cooling tower 110°. The pressure of water vapor of 80° temperature is 0.505 pounds per square inch. The pressure of the water vapor in the air is $0.428\times 0.70=0.30$ pounds per square inch. The pressure difference is therefore 0.20 pounds per square inch, and the rate of evaporation is $0.20\times 1.5=0.30$ pounds per square foot. The pressure corresponding to the temperature of the entering water is 1.27 pounds per square inch. Subtracting the 0.20 pounds difference in pressure to maintain the computed rate of evaporation we will have for the pressure of the water vapor in the air discharged, 1.07 pounds per square inch, which corresponds to a temperature of 104°. We may, in practice, assume that the air will come from the tower saturated with moisture at this temperature and neglect the superheat of the water vapor entering the tower. The weight of water vapor entering the tower per cubic foot of air supplied is $0.00135\times 0.70=0.00095$ and its total heat will be $1094.3\times 0.00095=1.04$ B.T.U. The pressure of the dry air entering the tower will be $14.70-0.30=14.40$. The pressure of the dry air leaving the tower will be $14.70-1.07=13.63$. The final volume of the air will be

$$\frac{14.40}{13.63}\times\frac{460+104}{460+75}=1.11 \text{ cu.ft.}$$

The quantity of moisture in the air leaving the tower will be $1.11\times 0.00319=0.00353$ pounds per cubic foot of air supplied, and the total heat will be $0.000353\times 1033.4=3.64$ B.T.U. The amount of heat carried away by the evaporation of the moisture will therefore be $3.64-1.04=2.6$ B.T.U. per cubic foot of air supplied. The air itself will be increased in temperature from 75° to 104°, and the sensible heat carried away will be (since a cubic foot of dry air weighs approximately 0.075 pounds) $0.075\times 29\times 0.238=0.52$ B.T.U. The heat carried away per cubic foot of air supplied will then be $0.52+2.6=3.12$ B.T.U. Assuming that the engine rejects 15,000 B.T.U. per horse-power per hour, we will then have for the required air supply,

$$\frac{15{,}000}{3.12}=4800 \text{ cu.ft.}$$

per hour or 80 cubic feet per minute. The quantity of water evaporated by each cubic foot of air supplied is $0.00352-0.00095=0.00257$ pounds. Hence the quantity of water evaporated per horse-power per hour will be $4800\times 0.00257=12.4$ pounds. We have already determined that the evaporation per square foot of checker work will be

0.30 pounds. Consequently the number square feet of checker work required per horse-power will be

$$\frac{12.4}{0.30}=41.$$

If this checker work be assumed to consist, as it often does, of 1-inch cypress planks laid up in such a way as to make a series of vertical flues 4 inches square, as shown in Fig. 143, we will have an evaporative surface of 7.7 square feet per cubic foot of checker work. This will give us about 5.3 cubic feet of checker work per horse-power. In the case of a 1000 horse-power cooling tower we would have a tower 14 feet square with a depth of checker work of about 28 feet.

Under these circumstances, the net area of the air passages will be about 125 square feet, and since the total quantity of air required is 80,000 cubic feet per minute, the velocity of the air in the passages will be 10.5 feet per second. The pressure required to produce this velocity in a tower of this height is about the pressure produced by a column of water $\frac{3}{8}$ inches high. The resistance offered by such a tower to the passage of the air varies directly as the depth of the checker work and as the 1.8 power of the velocity of the air. Since it is impracticable in the case of a natural-draft tower, to obtain by means of the difference in density of the air within and without the tower a difference in pressure as great as the figure given, a natural-draft tower will have a less depth of checker work and a much larger ground area for the same capacity. In general, on account of the lower velocity of the air, the rate of evaporation in a natural-draft tower per square foot of checker work will be about 60 per cent of the rate in a forced-draft tower, consequently, about $1\frac{2}{3}$ times as much evaporative area must be provided as would be provided in the case of a forced draft tower. In order to produce the required draft, a shaft or chimney extends from 50 to 100 feet above the top of the checker work on a natural-draft tower. The draft which such a tower will produce may be found by finding the difference in weight of a cubic foot of external air and a cubic foot of the air coming from the checker work and multiplying this by the vertical distance from the top of the checker work to the top of the tower. The product will be the difference in pressure in pounds per square foot, which may be reduced to inches of water by multiplying by 0.192.

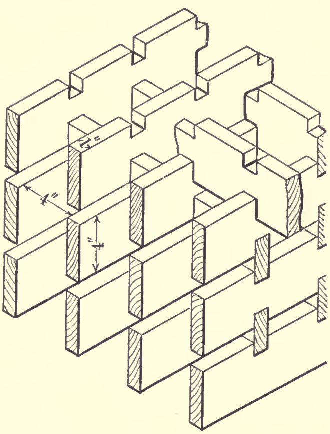


Fig. 143.—Arrangement of checker work for cooling tower.

A natural-draft tower of the same capacity as the one we have just designed would have 8 cubic feet of checker work per horse-power, or 8000 cubic feet altogether. The weight of the external air per cubic foot will be

$$\frac{14.40\times144}{53.3\times535}+0.00093=0.0740 \text{ lbs.}$$

The weight per cubic foot of the air in the shaft will be

$$\frac{13.63\times144}{53.3\times564}+.00319=.0687.$$

The difference in density will be 0.0053, and the draft produced will be 0.001 inches of water per foot in height of the shaft.

Assuming the same depth of checker work as in the forced-draft tower, the ground area will be 1.66 times, and the air velocity only 0.60 times the former value. The resistance of the checker work will therefore be $\frac{3}{8}\times 0.60^{1.8} = 0.15$ inches of water. The height of tower required would be 150 feet, which is excessive. Since the resistance of the tower for a given air velocity is proportional to the depth of checker work, and the velocity is inversely proportional to this depth, the height of the shaft is proportional to the 2.8 power of the depth of the checker work. Assuming a reasonable height of shaft, say, 60 feet, we will have for the depth of the checker work

$$28^{2.8}\frac{60}{150} = 26 \text{ ft.}$$

The area of the base of the tower will then be

$$\frac{8000}{26} = 286 \text{ sq.ft.},$$

or the tower will be then 17 feet square.

The volume of the checker work and the draft required to maintain the required air velocity varies greatly with the form of the checker work. The draft needed can be computed only from the observed performances of similar towers. In forced draft towers the power required by the fans is about one per cent of the power of the station.

PROBLEMS

1. It is desired to maintain the temperature of a cooling pond at 75°. The mean summer temperature is 65° and the humidity 50 per cent. It is used in connection with a 200 horse-power station operating 12 hours per day, using 18 lbs. of water per horse-power per hour. Find the area of cooling pond required. Ans. 21,800 sq.ft.

2. What area of spray pond would be required for the above plant? Ans. 600 ft.

3. One cubic foot of air enters a cooling tower at a temperature of 80° and a humidity of 80 per cent. Find the total heat present in the water vapor, disregarding the superheat of the vapor. Ans. 1.375 B.T.U.

4. Within the tower this cubic foot of air is raised to a temperature of 110° and saturated with moisture. Find its final volume. Ans. 1.122 cu.ft.

5. Find the total heat of the water vapor contained in this air. Ans. 4.680 B.T.U.

6. Find the increase in the sensible heat of the air. Ans. 0.52 B.T.U.

7. Find the heat carried away per cubic foot of air supplied. Ans. 4.08 B.T.U.

8. Find the heat carried away per pound of water evaporated. Ans. 1.370 B.T.U.

9. Water comes from the above cooling tower at a temperature of 95°. What is the rate of evaporation per square foot of checker work, assuming forced draft. Ans. 0.615 lbs.

10. Find the number of square feet of checker work required for a power plant of 1000 horse-power, rejecting 12,000 B.T.U. per horse-power hour. Ans. 14,200 sq.ft.

11. How many cubic feet of air will be required per hour by the above plant? Ans. 2,940,000 cu.ft.

CHAPTER XVIII

HOT AIR ENGINES

276. Characteristics of the Hot Air Engine. The hot air engine is a heat engine which uses air or other permanent gas as a working fluid. Since there is no advantage gained by employing any other gas, air is the working fluid invariably chosen. The hot air engine may be distinguished from the internal combustion engine by the manner in which the working fluid is heated, namely by the conduction of heat from some external source and not by the combustion of the working fluid itself. It is, therefore, unnecessary for the hot air engine to reject the working fluid and take a fresh supply at the completion of each cycle, as is done in the case of the internal combustion engine. The hot air engine, since the advent of the internal combustion engine, is not of great commercial importance. However, several of these engines afford excellent illustrations of important principles which are of great interest in connection with the probable development of internal combustion engines and refrigeration machinery, and are therefore worthy of careful study.

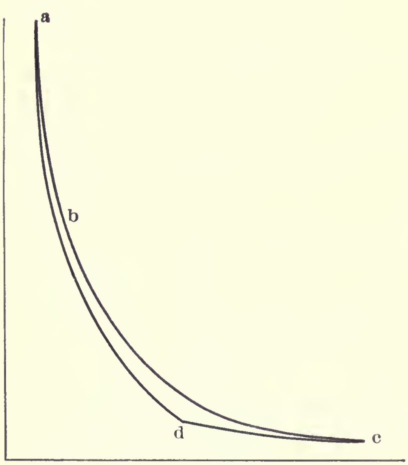


FIG. 144.—Theoretical card, to scale, from a Carnot cycle air engine.

277. The Carnot Air Engine. The most efficient, and in theory the simplest cycle which may be performed by the working fluid of a hot air engine, is the Carnot cycle. The Carnot cycle has never been used in any practical engine for the reason that the cylinder volume and the pressure range required in order to produce a very moderate amount of power, are very great. This may be seen by reference to Fig. 144, which shows to scale the Watt diagram of the Carnot cycle for air for the temperature range from 70° to 800° F. It will be noted that the card is extremely "thin," although the pressure range is very large.

This is a condition of affairs which makes for very low mechanical efficiency and multiplies greatly the practical difficulties of operation.

Assume that the Carnot cycle engine whose Watt diagram is illustrated in Fig. 144 uses 1 pound of air for its working fluid at an initial temperature (T_c) of 530° absolute, an initial pressure (P_c) of 2000 pounds per square foot (i.e., 13.9 pounds per square inch) and an initial volume (V_c) of

$$V_c = \frac{W\ R\ T_c}{P_c} = \frac{1 \times 53.3 \times 530}{2000} = 14.2 \text{ cu.ft.}$$

Assume that this air is compressed isothermally until its volume at d (V_d) is 7.1 cubic feet and its pressure (P_d) is 4000 pounds per square foot. Next, the air will be compressed adiabatically from d to a, until its temperature (T_a) is 1060° absolute. The pressure (P_a) will be

$$P_a = P_d\left(\frac{T_a}{T_d}\right)^{\frac{\gamma}{\gamma-1}} = 4000\left(\frac{1060}{530}\right)^{\frac{1.4}{.4}} = 45{,}000 \text{ lbs. per sq.ft.}$$

The volume of the air will of course be

$$V_a = V_d\left(\frac{T_a}{T_d}\right)^{\frac{1}{\gamma-1}} = 7.1\left(\frac{530}{1060}\right)^{2.5} = 1.26 \text{ cu.ft.}$$

The gas will now expand, the ratio of isothermal expansion being the same as the ratio of isothermal compression, namely two to one, and the volume and pressure at b will become 2.52 cubic feet and 22,700 pounds per square foot, respectively. The efficiency of the cycle is

$$\frac{T_a - T_c}{T_a} = \frac{1060 - 530}{1060} = 50\%,$$

which is about the theoretical efficiency of the internal combustion engine cycles usually employed. The net work done during the cycle is 50 per cent of the mechanical equivalent of the heat supplied during isothermal expansion, and is $\frac{1}{2}P_1\ V_1 \log_e r = \frac{1}{2} \times 45{,}000 \times 1.26 \times \log_e 2 = 39{,}200$ ft.lbs. Dividing this by the swept volume, which is $V_v, - V_a$ or 12.9 cubic feet, we will have 3040 pounds per square foot for the mean effective pressure.

It will be seen from the above computations that the maximum pressure is 320 pounds per square inch, while the mean pressure is only 20 pounds per square inch, or less than $^1/_{15}$ of the maximum pressure. In the case of the steam engine, it is very seldom that the mean effective pressure falls below one-half or one-third of the maximum pressure. Since the friction loss in an engine of a given power is approximately proportional to the ratio of the maximum to the mean pressure, it may be seen that the friction loss in the case of a Carnot cycle hot air engine is from

five to eight times that of a steam engine of equal indicated power. Owing to this fact, no hot air engine operating on the Carnot cycle or any approximation to it has ever been successfully used.

278. The Joule Hot Air Engine. The simplest of the practical cycles employing hot air as the working fluid is the Joule cycle. The operation of the Joule cycle may be understood by reference to Fig. 145. In cylinder A, a quantity of air is compressed adiabatically to some high pressure and is then discharged at constant pressure into the heater chamber B. Cylinder C takes the same mass of air from the heater chamber and expands it down to its original pressure. The volume of the heating and cooling chambers is so great that no sensible variation in pressure occurs. After expansion, the gas is rejected to the cooling chamber d at constant pressure. Here its temperature is reduced to the initial temperature and it again enters cylinder A to repeat the cycle. It is not necessary that a cooling chamber be provided, as the working fluid may be rejected to the atmosphere and a fresh quantity taken from the atmosphere by cylinder A. The equivalent series of processes consist of first, compression at constant pressure; second, adiabatic compression; third, expansion at constant pressure; fourth, adiabatic expansion. The Watt diagram is illustrated in Fig. 146. Assume that the temperature of the air in

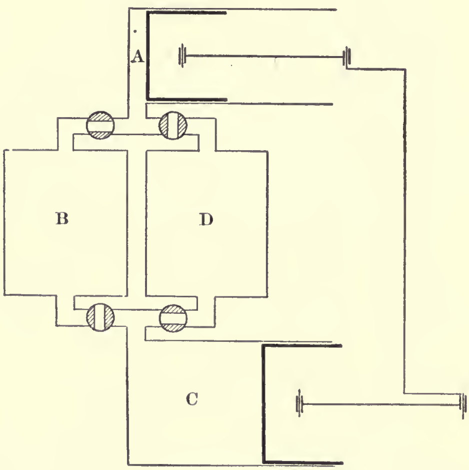


FIG. 145.—Diagram of a Joule cycle air engine.

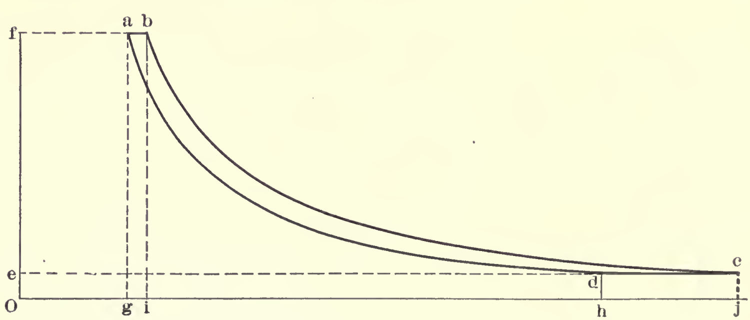


FIG. 146.—Watt diagram from a Joule cycle engine.

the heater chamber is 1200° absolute (740° F.) and the temperature of the atmosphere 530° absolute (70° F.), that the pressure in the heater chamber is 150 pounds absolute, per square inch, and that one pound of air is introduced into and withdrawn from the heater, per cycle. The temperature of compression (at a) may be found by the formula;

$$T_a = T_d\left(\frac{P_a}{P_d}\right)^{\frac{\gamma-1}{\gamma}},$$

or

$$T_b = 530\left(\frac{150}{14.7}\right) = 1030°,$$

which is the temperature of the air entering the heater chamber. The work of compression in cylinder A may be obtained by the formula:

$$U = R\frac{(T_2 - T_1)}{\gamma - 1} = 53.3\frac{(1030-530)}{0.4} = 66{,}700 \text{ foot pounds. (area } h\ d\ a\ g).$$

The work required to deliver the air from cylinder A into the heating chamber will be RT, which becomes $53.3 \times 1030 = 54{,}900$ foot-pounds (area $g\ a\ f\ o$). The heat imparted to the air in raising its temperature from 1030° to 1200° at constant pressure is

$$K_p\,(T_2 - T_1),$$

which is $185.5 \times 170 = 31{,}700$ foot-pounds. The work done by the air while entering the cylinder C is equal to $53.3 \times 1200 = 64{,}000$ foot-pounds. (area $f\ b\ i\ o$) The air is expanded from a pressure of 150 pounds absolute and an initial temperature of 1200° to a pressure of 14.7, and its final temperature is therefore

$$T_c = 1200\left(\frac{14.7}{150}\right)^{\cdot 286} = 618°,$$

The work of expansion will be

$$53.3\left(\frac{1200-618}{0.4}\right) = 77{,}500 \text{ ft.lbs.}$$

(area $b\ c\ j\ i$). The work done by the air in entering cylinder A from the cooler is $53.3 \times 530 = 28{,}300$ foot-pounds (area $e\ d\ h\ o$). The work done in expelling the air from cylinder C is $53.3 \times 618 = 33{,}000$ foot-pounds (area $e\ c\ j\ o$). The heat rejected in the cooling chamber will be

$$(618 - 530) \times 186.5 = 16{,}400 \text{ ft.lbs.}$$

The quantity of work done is the difference between the work done upon the air in compressing it and delivering it to the heater and rejecting it to the cooler and that done by the air in entering the cylinders and expanding in cylinder C, and is 15,300 foot-pounds (area $a\ b\ c\ d$). Dividing this by the mechanical equivalent of the heat imparted to the pound of air in the heater, we will have for the efficiency of the engine 48.2 per cent. An inspection of the above work will show that the heat supplied is proportional to $T_b - T_a$, the heat rejected to $T_c - T_d$, and the work done to $(T_b - T_a) - (T_c - T_d)$. The efficiency of the Joule cycle is therefore given by the expression

$$E = \frac{T_b + T_d - T_a - T_c}{T_b - T_a},$$

in which T_b = the temperature of the air leaving the heater, T_a = the temperature of the air entering the heater, T_c = the temperature of the air entering the cooler, or rejected from the engine, T_d = the temperature of the air leaving the cooler, or taken into the engine.

279. The Stirling Hot Air Engine. The Stirling engine utilizes the regenerator principle in order to attain the efficiency of the Carnot cycle without the accompanying mechanical disadvantages. The theory of this engine will be best understood by reference to Fig. 147 although the parts of the engine, in practice, are arranged in an entirely different manner. In the figure, A is a piston which works within the large cylinder B, C is a piston which works within the small cylinder D, which is connected with the large cylinder by the passage F. Assume that both pistons are at the lowest point of their stroke, as shown in the illustration. If piston A now be caused to rise, it will transfer the air above it from its upper to its under side, causing it to flow through the regenerator, marked R, which may consist of a large quantity of wire gauze. The air above the piston A is exposed to the action of a cooler, as shown, and is therefore at the temperature of the cooler, T_c. The air below the piston A is exposed

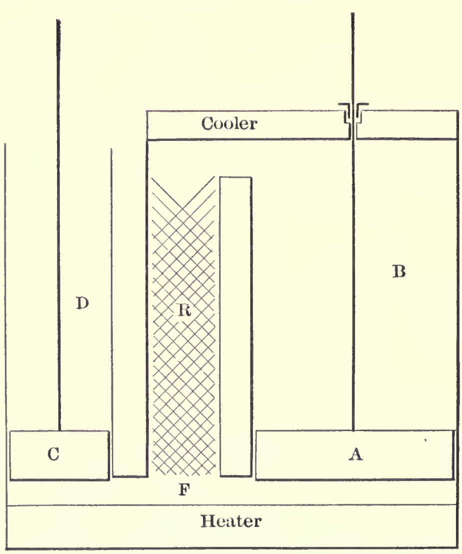


FIG. 147.—Diagram of a Stirling air engine.

to the fire, or to some other source of heat, and is therefore at the temperature of the heater, T_f. In passing through the regenerator, the temperature of the air will be raised from the temperature of the cooler to the temperature of the heater, by the regenerative action. As a result of this transfer and increase in temperature of the air, its pressure will rise. When piston A has reached the top of its stroke, the pressure of the air will have risen from its original value P_c, to the value P_f. Piston C is now permitted to rise, the air expanding isothermally, absorbing heat and performing work during the process. Piston C having reached the top of its stroke, piston A is now depressed, forcing the air below it through the regenerator and past the cooler into the upper part of B. As it passes through the regenerator, the air is cooled from the temperature of the heater to the temperature of the cooler. Piston C is now forced downward, isothermally compressing the air, while it rejects heat to the cooler. No work is done by or upon piston B, and the net work of the cycle is the difference between the work performed upon piston D during its upward stroke, and by it during its downward stroke.

The efficiency of a Stirling engine is in theory the same as the efficiency of a Carnot cycle engine. In practice, of course, it is necessary that there be a considerable temperature difference between the heater and the air under the transfer piston, between the cooler and the air above the transfer piston, and between any point in the regenerator and the air passing that point. The card given by the working cylinder of the engine is shown in Fig. 148. Line $a\ b$ represents the expansion of the air during the rise of piston C. Line $b\ c$ represents the fall in pressure of the air at constant volume (i.e., while the piston C is at the top of its stroke) on account of its transfer by piston A. Line $c\ d$ represents the compression of the air while piston C descends. Line $d\ a$ represents the rise in pressure at constant volume, while the transfer piston is transferring the air from the upper to the lower side.

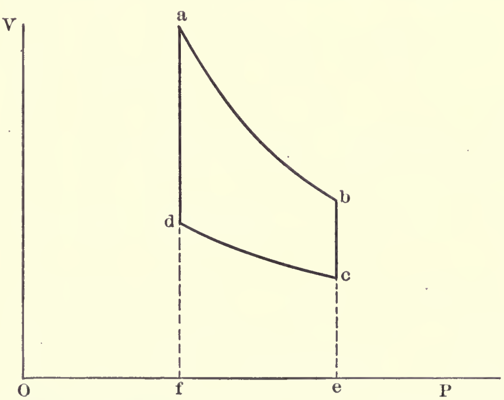


Fig. 148.—Watt diagram from a Stirling engine.

Assuming that 1 pound of air having a pressure of 100 pounds per square inch absolute at point d, is used as the working fluid, that the ratio of isothermal expansion is 2, that the temperature of the air above the transfer piston is 70° F., and that below the transfer system piston 740° F., and neglecting the volume of the regenerator

spaces and air passages, we will obtain the following results from the Stirling cycle. During the rise of the transfer piston, no work will be done or absorbed, the temperature of the air will rise from 530° absolute to 1200° absolute and the pressure will rise from 100 to 226.5 pounds per square inch. The air is heated by heat stored in the regenerator, and during this portion of the cycle it receives no heat from the heater. During the rise of piston D, the air expands isothermally, receiving heat from the heater in order to maintain its temperature constant. The amount of heat so received is equal to the amount of work done by the air upon piston C, which is $53.34\times1200\times\log_e 2=$ 44,350 foot-pounds. This is the mechanical equivalent of the heat received by the air from the heater, during this portion of the cycle, and is represented by the area $a\,b\,e\,f$. At the end of isothermal expansion, the pressure of the air has fallen to 113.25 pounds per square inch. During the descent of the transfer piston, a part of the air is transferred through the regenerator, and its temperature falls to that of the cooler. No work is done or absorbed during this portion of the cycle and no heat is taken from the heater or imparted to the cooler. Since the ratio of expansion is 2 to 1, the volumes of the transfer cylinder and the working cylinders are equal and are each

$$\frac{530\times53.34}{100\times144}=1.964\text{ cu.ft.}$$

Hence after the transfer of part of the air to the upper side of piston B has been effected we will have 1.964 cubic feet of air at a temperature of 530° and 1.964 cubic feet at a temperature of 1200°. Since the pressures and the volumes are the same in each case, the masses will be inversely proportional to the absolute temperatures, and the quantity in the transfer cylinder will be

$$\frac{1200}{1200+530}=0.694\text{ lbs.}$$

The pressure of this air (P_c) will be

$$\frac{0.694\times53.34\times530}{144\times1.964}=69.4\text{ lbs. per square inch.}$$

During the descent of piston D, the remainder of the air which is contained in cylinder C, is transferred through the regenerator and the cooler to the upper side of the transfer piston, and the whole quantity of air is compressed. That portion of the air within cylinder C which is continually diminishing in quantity rejects to the heater the heat equivalent of the work performed in compressing it. The air in the transfer cylinder, which is continually increasing in quantity, rejects to the cooler the heat equivalent of the work spent in compressing it. In order to find the amount of work done during the compression, we must determine the relation between the pressure and volume on the whole mass of gas contained in the engine during the descent of the working piston. Let V be the volume of the working cylinder. Then the mass of the air in the working cylinder, W_1, will be to the mass of the air in the transfer cylinder, W_2, directly as the volume of the air in the working cylinder is to that in the transfer cylinder and inversely as the temperature of the air in the working cylinder is to that in the transfer cylinder. Consequently, we may write

$$W_1=\frac{530\,V}{530\,V+1200\times1.966}.$$

The pressure of this mass may be obtained from the formula $PV = W_1 R T$ and will be

$$P = \frac{53.34 \times 1200 \times 530}{530\ V + 2360} = \frac{63{,}900}{V + 445}.$$

The work done upon the gas will therefore be

$$\int_{V_c}^{V_d} P\, dV = 63{,}900 \int_0^{1.966} \frac{dV}{V + 4.45}.$$

Integrating, we obtain

$$63{,}900 \log_e \frac{6.416}{4.450} = 23{,}400 \text{ ft.-lbs.}$$

Subtracting this from 44,350 foot-pounds the work done by the gas during the rise of the working piston, we will have 20,950 foot-pounds for the net work of the cycle. The swept volume is 1.964 cubic feet and the mean effective pressure is

$$\frac{20{,}950}{144 \times 1.966} = 74.0 \text{ lbs. per square inch.}$$

The ratio of the maximum to the mean effective pressure is $\frac{226.5}{74.0}$, or about 3 to 1, a condition of affairs very much more favorable to mechanical efficiency than is the case when the Carnot cycle is employed. The quantity of heat rejected to the cooler is equal to the work done in compressing the variable quantity of gas contained above the transfer piston during the descent of the working piston. Since V equals the volume of gas in the working cylinder during this period, the total volume of the gas will be $V + 1.966$. The amount of work done during any small portion of the stroke of the working piston upon the two quantities of gas will be proportional to their volumes at that instant. Consequently, by multiplying the total work done during any instant by the ratio of the volume of the transfer cylinder to the total volume of the gas at that instant we will have the work done upon the gas in the transfer cylinder during that instant. Multiplying the equation for the work performed upon the gas during the descent of the working piston by $\frac{1.966}{V + 1.966}$, we will have

$$1.966 \times 63{,}900 \int_0^{1.966} \frac{dV}{(V + 4.45)(V + 1.966)}.$$

This may be written

$$125{,}600 \int_0^{1.966} \frac{dV}{(8.74 + 6.416\ V + V^2)}.$$

Integrating this, we will have

$$125{,}600 \left(\frac{1}{41.10 - 4 \times 8.74}\right) \log_e \left(\frac{2V + 6.416 - 41.1 - 4 \times 8.74}{2V + 6.416 + 41.1 - 4 \times 8.74}\right)_0^{1.966}.$$

Solving this we will have for the mechanical equivalent of the heat rejected to the cooler 16,540 foot-pounds. Subtracting this quantity from the total work done upon the gas in compressing it we will have the mechanical equivalent of the heat restored to the heater, which is 6,860 foot-pounds. In order to obtain the efficiency of the cycle, we must divide the net work of the cycle by the net heat supplied, which

is equal to the heat supplied during isothermal expansion less the heat restored to the heater during compression, and we will obtain

$$\frac{20{,}950}{44{,}350-6860}=55.8 \text{ per cent.}$$

It will be noted that the cycle is composed exclusively of reversible processes, and the efficiency of the cycle is in consequence that of the Carnot engine, and is, for the case chosen,

$$\frac{1200-530}{1200}=55.8 \text{ per cent,}$$

which is the same as was obtained by computation of the work performed, and the heat supplied and rejected.

Had the working cylinder been connected with the upper end of the transfer cylinder the cycle would have differed from that described, in that during the descent of the working piston the compression would have been isothermal, and during the rise of the working piston, heat would have been absorbed from both the heater and the cooler. The efficiency of the cycle would be exactly the same as before.

In practice, the Stirling engine is subject to several losses, and the card given by the engine is not exactly of the form computed, since the regenerator and the connecting passages have some volume. Neither is the action of the regenerator a perfectly reversible process in practice. Usually, the temperature difference between the air entering and that coming from the cool end of the regenerator is from 5 to 20 per cent of the difference in temperature of the two ends of the regenerator. On this account, the actual efficiency of the Stirling engine is only about 60 per cent of its theoretical efficiency. In addition, there are practical difficulties encountered in its use, which may, however, be overcome by the use of proper materials. Modifications of the Stirling cycle will probably serve as a basis for future improvements in the internal combustion engine, since this engine is in theory the most efficient one which has ever been practically successful.

280. The Ericsson Hot Air Engine. The principle of the Ericsson hot air engine is shown in Fig. 149. This engine is usually built only in small sizes and used for pumping water. The method of operation is as follows: Within the cylinder A is a gas-tight piston B, termed the working piston. Through a stuffing-box in this piston there passes a rod which operates the loose-fitting plunger, C. The purpose of this plunger is to transfer the air from the lower to the upper end of the cylinder and back again, and it is therefore termed the transfer plunger. Both piston and plunger being at the top of the stroke, as is shown in Fig. 149, the plunger descends, transferring the air from the furnace at the bottom to the comparatively cool region at the top. In its passage

the air flows in a thin sheet over the water-cooled surface of the cylinder, and its heat is transferred to the water jacket. The air being cooled, its pressure is reduced. While the transfer plunger remains at the bottom of its stroke, the working piston descends, compressing the air contained in the cylinder. The transfer plunger now rises while the piston remains stationary, transferring the air to the lower end of the cylinder, where it is heated, with resulting increase of pressure. The piston now rises as a result of the increase in pressure and performs work.

Since this engine lacks a regenerator, it is less efficient than the Stirling engine. Its principal merit is that it is very simple and unlikely to get

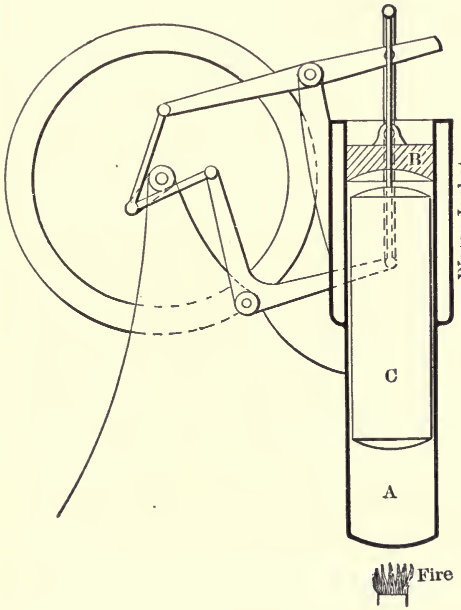


FIG. 149.

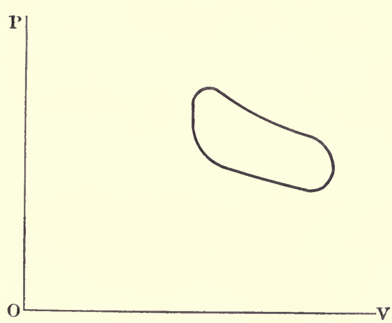


FIG. 150

out of order. In practice, the piston and plunger are so connected to the crank of the engine that they both are in motion continually, instead of each one stopping while the other is performing its stroke. This is accomplished by the mechanism shown in Fig. 149, the piston and water pump being operated by the walking-beam, while the transfer plunger is operated by the bell crank. It will be seen that the plunger is near the end of its stroke and is almost motionless while the piston has its maximum velocity and is at the middle of its stroke. The form of card given by the Ericsson engine is shown in Fig. 150.

PROBLEMS

1. Find the diameter and length of stroke which would theoretically be required by an engine utilizing the Carnot cycle worked out in Art. 277. Assume a speed of 150 revolutions per minute, that the length of stroke is $1\frac{1}{2}$ times the diameter, that the engine is 100 horse-power and is single acting. Ans. $22'' \times 33''$.

2. Find the length of stroke and the diameters of the compression and working cylinders of an engine utilizing the Joule cycle worked out in Art. 278. The engine is of 100 horse-power and makes 150 revolutions per minute. Assume the length of stroke to be equal to $1\frac{1}{2}$ times the diameter of the compression cylinder and that the cylinders are single acting. Ans. $30.5'' \times 45.7''$ and $32.9'' \times 45.7''$.

3. A Stirling engine operates between temperatures of 200 and 1000° F. The pressure when both pistons are in their lowest positions is 150 lbs. per sq.in. Find the pressure after the transfer cylinder is raised. Ans. 332 lbs. per sq.in.

4. Assuming 1 lb. of air as the working fluid and that the volume of the transfer cylinder is twice the volume of the working cylinder, find the work of isothermal expansion. Ans. 31,550 ft.lbs.

7. Find the pressure ta the end of isothe*r*mal expansion. Ans. 221 lb per sq.in.

6. Find the pressure of the air after the descent of the transfer piston.

Ans. 122 lbs. per sq.in.

7. Find the value of the index of the compression curve, assuming that it follows the law

$$PV^n = \text{a constant}.$$

Ans. $n = .51$

8. Find the work of compression on the same assumption. Ans. 16100 ft.lbs.

9. Find the net work of the cycle. Ans. 15450 ft.lbs.

10. Find the net work per cubic foot of swept volume of the working cylinder.

Ans. 18,940 ft.lbs.

11. Find the size of working cylinder required for a 100 horse-power engine, operating at 150 revolutions per minute, utilizing the Stirling cycle just developed. Make the stroke $1\frac{1}{2}$ times the diameter of the cylinder. Ans. $11.95'' \times 17.9''$.

CHAPTER XIX

THE INTERNAL COMBUSTION ENGINE

281. Characteristics of the Internal Combustion Engine. An internal combustion or gas engine is a heat engine in which the working fluid consists of a mixture of air and inflammable gas or vapor, the combustion of which furnishes the heat necessary for the operation of the mechanism. The internal combustion engine differs from all other heat engines in that the combustion which supplies the heat occurs within the working chamber or cylinder of the engine itself. In all other heat engines, the working fluid and the combustible are separate substances, and the working fluid is usually heated in a separate chamber from that in which it performs its work. As in any heat engine, the working fluid of the internal combustion engine is caused to perform a thermodynamic cycle whose form is determined by the nature of the fluid and the arrangement of the engine mechanism.

282. The Otto Cycle Engine. The internal combustion engine cycle which is in most common use is usually termed the **Otto cycle,** and was first proposed by Beau de Rochas. It is also known as the **constant-volume cycle,** and as the **four-stroke cycle.** The operations of the Otto cycle may be understood by reference to Fig. 151. The engine cylinder is an iron

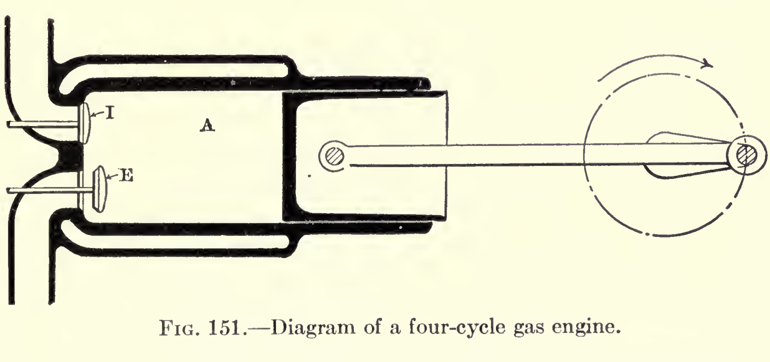


FIG. 151.—Diagram of a four-cycle gas engine.

casting *A* provided with an **inlet valve I** and an **exhaust valve E.** Within the cylinder moves a gas-tight piston which is often made of the form shown, performing at the same time the functions of a piston and of a cross-head. This type of piston is known as a **trunk piston.** As the

crank revolves in the direction shown by the arrow, the piston is moved back and forth. Assume that the engine is in the position shown, with the crank on the outer center, then as the crank revolves it will begin to force the piston inward. The valve E being held open by the mechanism of the engine during this stroke, the contents of the cylinder will be expelled. This first stroke is therefore termed the **exhaust stroke** of the cycle. When the piston has reached the end of its exhaust stroke, the valve E closes and the valve I opens. The piston then begins to move forward, and draws in a quantity of air with which is mixed a combustible gas. This second stroke is known as the **suction stroke** of the cycle, and the mixture drawn into the cylinder is termed the **charge.** When the piston reaches the end of this stroke, the valve I also closes, and the crank continuing to revolve, the piston is again forced inward, adiabatically compressing the charge into the **clearance space** at the end of the cylinder. This third stroke is known as the **compression stroke.** The volume of the clearance space, which usually ranges from 12 per cent to 30 per cent of the swept volume of the cylinder, is termed the **clearance volume.** When the piston reaches the end of the compression stroke, the charge, which is now highly compressed and heated, is ignited, usually by means of an electric spark. On account of its high temperature and pressure the charge burns almost instantly, and the heat so generated, by raising the temperature of the charge, very greatly increases its pressure. During the fourth stroke of the piston the charge expands adiabatically. Because of the great pressure resulting from its explosion much more work will be done by the charge during this stroke than was done upon it during the compression stroke. The fourth stroke is therefore known as the **working stroke** of the cycle. When the piston reaches the end of the working stroke, the exhaust valve E again opens, and both the working fluid and the mechanism of the engine return to their original condition.

It will be seen that it requires four strokes of the engine to complete the cycle. During the first or exhaust stroke the pressure upon the piston is only slightly above that of the atmosphere and during the second or suction stroke only slightly below. Although the amount of work performed in expelling the burned charge and in drawing in a fresh one varies somewhat with the speed of the engine and the size of the valves and gas passages, it is very small under ordinary conditions. Hence the suction and exhaust strokes need not be considered in connection with the thermodynamic cycle, which is performed during the compression and working strokes only.

283. The States of the Working Fluid During the Otto Cycle. The pressure-volume diagram of the working fluid of an Otto cycle engine is shown in Fig. 152. The horizontal distance Oc' represents the clearance volume, while the distance $c'a'$ represents the swept volume of the cylin-

der. $a\ c$ is the adiabatic compression line, the cylinder containing a charge at atmospheric temperature and pressure at point a. At point c, the charge is instantly heated at constant volume by the explosion, the pressure rising as represented by line cx. Line xt is the adiabatic expansion line of the charge and the line $t\ a$ represents the cooling of this charge at constant volume at the end of expansion. The computations of the

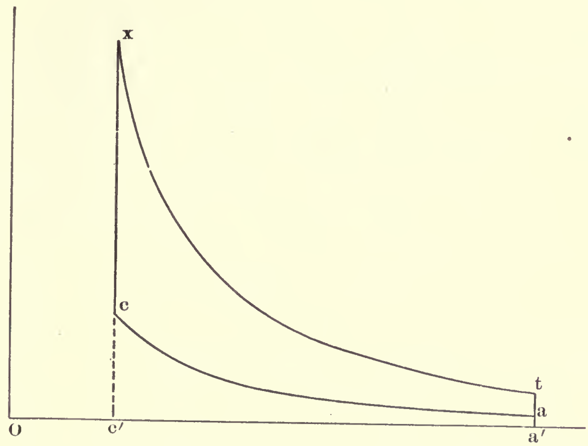


FIG. 152.—Theoretical card from an Otto cycle engine.

pressures, temperatures, and so on of the working fluid at various points in the cycle may be performed as follows:

Let P_a = the absolute pressure of the atmosphere in pounds per square foot;

V_a = the swept volume + V_c in cubic feet;

T_a = the absolute temperature of the atmosphere;

P_c = the absolute pressure of compression in pounds per square foot;

V_c = the volume of the compression space in cubic feet;

T_c = the absolute temperature of compression;

P_x = the absolute pressure of explosion;

$V_x = V_c$ = the volume at explosion;

T_x = the absolute temperature of explosion;

P_t = the absolute terminal pressure in pounds per square foot;

$V_t = V_a$ the terminal volume in cubic feet;

T_t = the absolute terminal temperature;

H_a = the heat in B.T.U.'s added at explosion;

H_r = the heat in B.T.U.'s rejected at exhaust;

W = the weight of gas contained in the cylinder;

$V_a - V_c$ = the swept volume in cubic feet $= V_t - V_x$.

The weight of the charge will be

$$W = \frac{P_a V_a}{R T_a} \text{ lbs.} \qquad (1)$$

The volume of the compression space will be

$$V_c = V_a \left(\frac{P_a}{P_c}\right)^{\frac{1}{\gamma}}. \qquad (2)$$

The temperature of compression will be

$$T_c = T_a \left(\frac{P_c}{P_a}\right)^{\frac{\gamma-1}{\gamma}} = T_a \left(\frac{V_a}{V_c}\right)^{\gamma-1}. \qquad (3)$$

The rise in temperature resulting from the explosion will be

$$T_x - T_c = \frac{H_a}{W C_v}. \qquad (4)$$

The heat added as a result of the explosion will be

$$H_a = W C_v (T_x - T_c) \qquad (5)$$

The terminal temperature will be

$$T_t = T_x \left(\frac{V_a}{V_c}\right)^{\gamma-1} = T_x \left(\frac{P_t}{P_x}\right)^{\frac{\gamma-1}{\gamma}}. \qquad (6)$$

It will also be

$$T_t = \frac{T_a T_x}{T_c}, \qquad (7)$$

since

$$T_c : T_a :: T_x : T_t. \qquad (8)$$

The fall in temperature at the end of expansion will be

$$T_t - T_a = \frac{H_r}{W C_v}. \qquad (9)$$

The heat rejected will be

$$H_r = W C_v (T_t - T_a). \qquad (10)$$

The efficiency of the cycle will be

$$E = \frac{H_a - H_r}{H_a}. \qquad (11)$$

or

$$E = \frac{C_v (T_x - T_c) - C_v (T_t - T_a)}{C_v (T_x - T_c)}. \qquad (12)$$

Simplifying, this becomes

$$E = 1 - \frac{T_t - T_a}{T_x - T_c}, \quad \ldots \ldots \ldots \quad (13)$$

which on substituting from (7) reduces to

$$E = 1 - \frac{T_a}{T_c}. \quad \ldots \ldots \ldots \quad (14)$$

It appears from the above equation that the higher the compression temperature (or pressure), the greater the efficiency of the cycle, and that the efficiency is theoretically independent of the explosion and terminal temperatures and pressures and of the amount of heat added. The relation between the compression pressure and the efficiency of the cycle may be seen in Fig. 153.

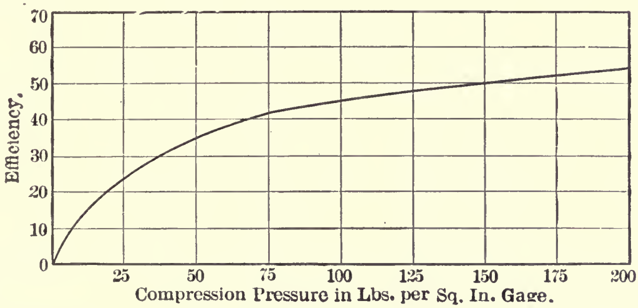


FIG. 153.—Relation between the efficiency and compression pressure.

The work of compression is

$$U_{ac} = \frac{P_c V_c - P_a V_a}{\gamma - 1} = \frac{R\,(T_c - T_a)}{\gamma - 1}. \quad \ldots \ldots \quad (15)$$

The work of expansion is

$$U_{xt} = \frac{P_x V_c - P_t V_a}{\gamma - 1} = \frac{R\,(T_x - T_t)}{\gamma - 1}. \quad \ldots \ldots \quad (16)$$

The work done is equal to

$$U = \frac{P_x V_c - P_t V_a - P_c V_a + P_a V_a}{\gamma - 1}$$

$$= \frac{(P_x - P_c)\,V_c - (P_t - P_a)\,V_a}{\gamma - 1}$$

$$= \frac{R}{\gamma - 1}\,(T_x + T_a - T_c - T_t). \quad \ldots \ldots \quad (17)$$

Net
The work done per cubic foot of swept volume is

$$\frac{U}{V_a - V_c}. \qquad \qquad (18)$$

The theoretical horse-power of an engine employing the cycle is

$$\text{H.P.} = \frac{U\,N}{33{,}000\,(V_a - V_c)}, \qquad \qquad (19)$$

in which N is the number of cycles (i.e., one-half of the number of revolutions) per minute.

Also for convenience in computation, it may be noted that

$$T_x = T_c + \frac{H_a}{C_v}, \qquad \qquad (20)$$

and

$$P_x : P_c :: P_t : P_a. \qquad \qquad (21)$$

The compression volume is generally expressed as a per cent of the swept volume thus

$$V_c\ (\text{in per cent}) = \frac{100\ V_c}{V_a - V_c}. \qquad \qquad (22)$$

It will be shown in the next chapter that the actual cycle performed in the Otto gas engine differs slightly from the theoretical cycle. The differences are not so great as they are in the case of steam engine cycles, however, and the actual card has very nearly the form of the theoretical card.

284. Example Showing the Method of Computing the States of the Working Fluid. The following example will serve to show the method of computing the form of the theoretical indicator card, the work done and the efficiency of an Otto cycle. Assume that the compression pressure is 140 pounds absolute (i.e., about 125 pounds gage), that the temperature of the gases after explosion will be 3500° absolute, and that the temperature of the mixture entering the cylinder is 70° F. The computations will be carried out for 1 pound of air. The initial volume is obtained from the equation

$$P\,V = W\,R\,T,$$

and is

$$V = \frac{1 \times 53.34 \times 530}{14.7 \times 144} = 13.32 \text{ cu.ft.} \qquad \qquad (1)$$

The volume of the compression space per pound of air will be

$$V_c = 13.32 \left(\frac{140}{14.7}\right)^{-\frac{1}{r}} = 2.69 \text{ cu.ft.} \qquad \qquad (2)$$

The swept volume per pound of air will be

$$13.32-2.69=10.63 \text{ cu.ft.} \quad \ldots \ldots \quad (3)$$

The temperature of compression will be

$$T_c=530\left(\frac{140}{14.7}\right)^{.29}=1020° \text{ absolute.} \quad \ldots \ldots \quad (4)$$

The pressure of explosion will be

$$P_x=\frac{3500}{1020}\times 140=480 \text{ lbs. absolute.} \quad \ldots \ldots \quad (5)$$

The efficiency of the cycle will be

$$1-\frac{530}{1020}=48 \text{ per cent.} \quad \ldots \ldots \quad (6)$$

The quantity of heat added at explosion will be

$$H_a=0.169(3500-1020)=419 \text{ B.T.U.} \quad \ldots \ldots \quad (7)$$

The work done per pound of air will be

$$419\times 0.480\times 777.5=156{,}200 \text{ ft.-lbs.} \quad \ldots \ldots \quad (8)$$

The pressure at release will be

$$P_t=\frac{14.7\times 140}{480}=42.9 \text{ lbs. per square inch.} \quad \ldots \ldots \quad (9)$$

The work done per pound of air may be checked in the following manner. The work of compression is

$$\frac{53.3\times(1020-530)}{.407}=64{,}200 \text{ ft.-lbs.} \quad \ldots \ldots \quad (10)$$

The work of expansion is

$$\frac{53.3\times\left(3500-530\frac{3500}{1020}\right)}{0.407}=220{,}200 \text{ ft.-lbs.} \quad \ldots \ldots \quad (11)$$

The difference is the net work of the cycle, or

$$220{,}200-64{,}200=156{,}000. \quad \ldots \ldots \quad (12)$$

The net work per cubic foot of swept volume per cycle is

$$\frac{156{,}000}{10.63}=14{,}650 \text{ ft.-lbs. per cubic foot.} \quad \ldots \ldots \quad (13)$$

The mean effective pressure shown by the card is

$$\frac{14{,}650}{144}=102 \text{ lbs. per square inch.} \quad \ldots \ldots \quad (14)$$

The horse-power theoretically developed by an engine operating on this cycle would be

$$\text{H.P.}=\frac{14{,}650\, N\,(V_a-V_c)}{33{,}000}, \quad \ldots \ldots \quad (15)$$

in which N is the number of cycles, or one-half the number of revolutions per minute.

The card for the engine may now be constructed by locating points P_aV_a, P_cV_c, P_xV_c, and P_tV_a, drawing perpendiculars between the first and fourth and the second

and third, and adiabatics between the first and second and third and fourth. This is the card illustrated in Fig. 152.

285. Methods of Governing the Otto Cycle Engine. Four methods are employed for controlling the speed and amount of power developed, in an Otto cycle engine. The first method is to cause the engine to miss explosions, thus reducing the number of working strokes which the engine makes in a given time. This is known as **hit-and-miss governing.** The second method is to throttle the port through which the charge enters the engine, reducing the weight of charge taken in and the power developed by its explosion. This is known as **throttle governing.** The third method is to reduce the proportion of combustible gas contained in the charge, and so to weaken the explosion. The fourth method is to delay the instant of ignition, and so to reduce the power developed by a given weight of charge.

If the power developed within the cylinder of an Otto cycle engine equipped with a hit-and-miss governor is greater than is necessary to keep the engine up to speed, the engine will gain speed. In order to prevent this, it is customary to so arrange the governing mechanism that it will prevent explosions so long as the engine is operating at more than normal speed. This may be accomplished in several ways. One method is to cause the governor to hold the exhaust valve open until the speed becomes normal. If the inlet valve operates automatically (i.e., if it is opened by the suction of the piston and not by some mechanical device), this will prevent the engine from sucking in a charge during the suction stroke, since it will cause it to take suction from the exhaust pipe, and no explosion will occur at the beginning of the next working stroke. A second method is to cause the governor mechanism to hold the inlet valve closed. A third method (which is, however, very seldom used) is to cause the governor mechanism to hold the inlet valve open during the compression stroke. When an engine is equipped with heavy fly-wheels and a sensitive governor, the hit-and-miss system gives fairly close regulation and very great economy. When extremely close speed regulation and uniform turning moment is unnecessary, it is the preferable method of speed regulation.

Throttling the mixture as it enters the engine cylinder gives a card of the form shown in Fig. 154. Line $a\ b$ is the exhaust stroke, line $b\ d$ is the suction stroke, line $d\ c$ the compression stroke and line $x\ t$ the working stroke. The dotted lines show the form of card which would be given were there no throttling. The area $d\ e\ b$ is the work required to suck the charge through the throttle, and is a loss. Since the compression pressure is reduced, the efficiency of the cycle is reduced somewhat, although this is partly counterbalanced by the lower terminal pressure resulting from the more complete expansion of the charge. This method of governing, therefore, reduces the efficiency of the engine. It is mechanically

satisfactory so long as the compression does not fall too low. When it does fall too low, on account of extreme throttling, ignition fails to take place, and other methods of regulation must be resorted to.

Throttling the inflammable component of the charge reduces the heat added at the instant of explosion, and therefore gives a card of the form shown in Fig. 155. The card shown in dotted lines represents the form

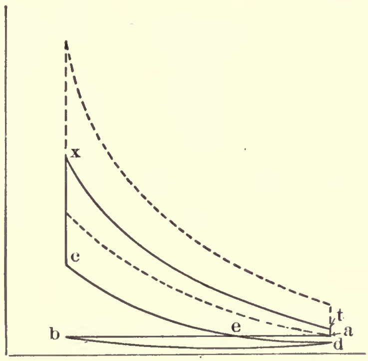


FIG. 154.—Card given by a throttle governed gas engine.

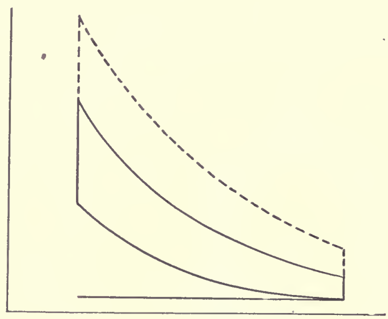


FIG. 155.—Card given when the gas is throttled.

of card given with the gas unthrottled. Theoretically this method of governing makes no reduction in the efficiency of the cycle. Practically its efficiency is less than that of the hit-and-miss method. As the mixture grows weaker it becomes more difficult to ignite, so that at low loads ignition fails.

A fourth method of governing a gas engine is to "delay the spark." This method is wasteful, since by retarding the ignition to some point in the working stroke, a portion of the power available is not utilized. The form of card produced may be seen in Fig. 156. The card shown in dotted lines is the one which would be given by the same charge were the spark properly advanced. It will be seen that a considerable amount of power is wasted. This method of governing is used in connection with throttle governing in operating automobile or other portable engines in which it is desired to vary the speed of the engine as well as the power. This method is there employed on account of its extreme convenience and adaptability, in spite of its inherent wastefulness.

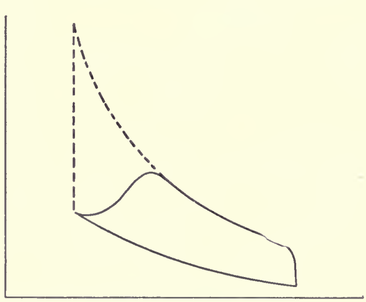


FIG. 156.—Result of delayed ignition.

Combinations of any of these methods of gas-engine governing may be used in the case of the Otto cycle engine. The usual method is to use hit-and-miss governing alone where a fairly constant speed is required in stationary work, to use hit-and-miss in connection with throttle governing where a constant speed and close regulation are required; and to use throttle governing in connection with delayed ignition where variable speed and power are required of the engine.

286. The Two-cycle Engine. The thermodynamic principle of the two-cycle engine is exactly the same as that of the Otto cycle engine. The details of the engine are, however, quite different. In Fig. 157 is shown a section of a two-cycle engine. The cylinder is an iron casting which is water jacketed in the manner shown. The crank case *b* is so arranged that it is practically gas tight. During the upward stroke of the piston *P*, a charge is drawn into the crank case through the check valve *V*. This charge consists of a mixture of air and combustible gas or vapor. When the piston descends, this charge is compressed within the crank case, its pressure rising to from 7 to 12 pounds gage, depending upon the volume of the crank case. When it reaches the bottom of its stroke, the piston uncovers two ports, one on each side of the cylinder. The port at the left marked *E* is the exhaust port, while that at the right marked *I* is the inlet port. Since the inlet port connects the crank case with the cylinder, the charge in the crank case will be forced into the cylinder on account of the difference in pressure. The form of the top of the piston is such that this charge is directed upward and into the cylinder, and as it enters, it expels the gases contained within the cylinder through the exhaust port. During the upward stroke of the piston, the charge now contained within the cylinder is adiabatically compressed. When the piston reaches the top of its stroke, the charge is ignited by the spark plug marked *S* and explodes. The piston descends during the working strokeof the engine. During its upward or compression stroke it has sucked a fresh charge into the crank case and during its downward or working stroke it compresses this charge. As the piston reaches the end of its stroke, the exhaust port is uncovered and the burned charge escapes from the cylinder on account of its pressure. An instant later the inlet port (which is nearer the

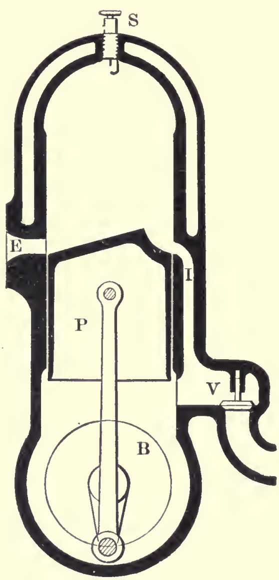


Fig. 157.—Section of a two-cycle engine.

bottom of the cylinder than the exhaust port) is uncovered, and the crank case compression forces the fresh charge into the cylinder. It will thus be seen that the engine makes one compression stroke and one working stroke each revolution.

It is usual in small two-cycle motors to employ the crank case as a pump in the manner already described. In large two-cycle engines, separate pumps are provided for compressing the gas and the air, and the engine is sometimes made double-acting in the manner shown in Fig. 158. In this engine the pumps deliver the charge to the working cylinder through the valve *I*, while the working piston *P* is in the position shown. The entering charge expels the spent charge through the ports marked *P*. As the working piston moves to the right it compresses the fresh charge, which is exploded when the piston reaches the end of its stroke. When it moves to the left, the right-hand charge is performing its working

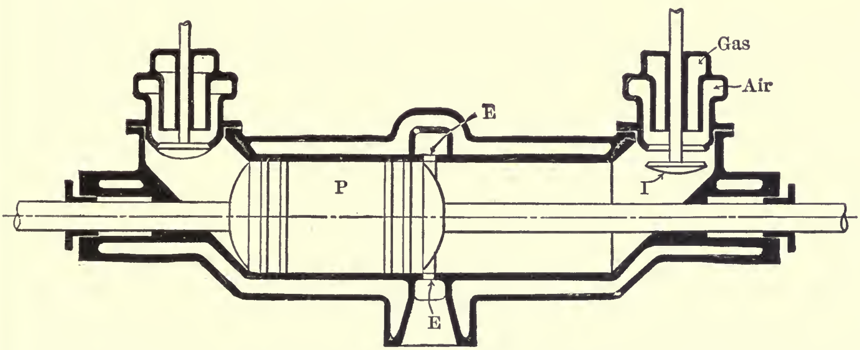


FIG. 158.—Section through the working cylinder of a Koerting two-cycle double acting engine.

stroke while the left hand charge is being compressed. The length of the piston is such that the single set of exhaust ports serves for both ends of the cylinder.

The advantages of the two-cycle over the four-cycle engine are that it requires a less number of valves and that it makes twice as many working strokes in a given number of revolutions. At low speed, a two-cycle engine gives nearly twice the power of a four-cycle engine of the same speed and size. At high speeds, however, a part of the fresh charge is apt to escape from the cylinder and a part of the spent charge is apt to remain. This results in a serious reduction in the power and efficiency of the two-cycle engine. The theoretical efficiency of the two-stroke cycle is, however, exactly the same as that of the four-stroke cycle having the same compression pressure, and the pressures, temperatures, volumes, and work done are computed in exactly the same manner.

287. The Sargent Cycle Engine. The Sargent cycle has been developed in order to provide a cycle having a greater efficiency than the Otto cycle, and also to provide better means of speed regulation for engines of large power. The theoretical card of a Sargent cycle engine is illustrated in Fig. 159. In the Sargent cycle, the admission of the charge is stopped at some point during the suction stroke. This point is marked A on the card. During the remainder of the stroke the charge is expanded below atmospheric pressure, its final pressure and volume being represented by the point n. During compression stroke the charge is compressed to the point C. Explosion then occurs, the pressure rising to point X. Adiabatic expansion then follows, the charge expanding to point L. The exhaust valve then opens and the pressure falls to that of the atmos-

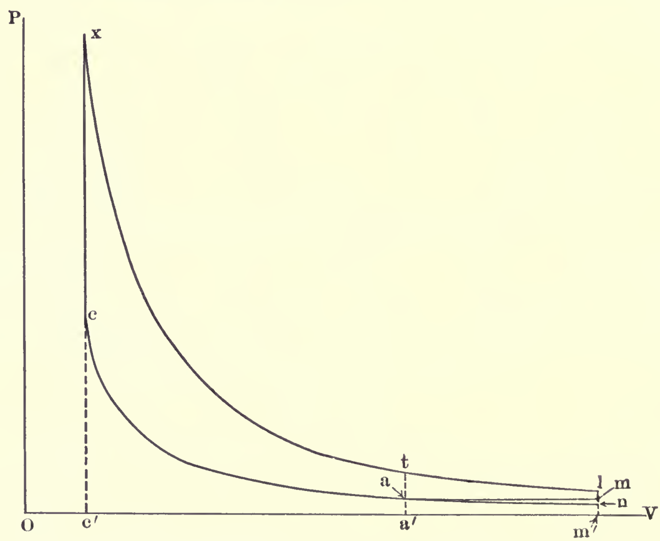


FIG. 159.—Theoretical card from a Sargent cycle engine.

phere, represented by point m. The spent charge is now expelled from the cylinder. A fresh charge is then drawn in during the early part of the suction stroke and the cycle is repeated. The amount of power developed is adjusted by causing the governor to close the inlet valve at the proper point in the suction stroke, the valve closing early when the power required is small, and remaining open until the end of the stroke when the maximum power is required of the engine.

It will be seen from an inspection of the Sargent cycle card that it consists of two parts, $a\ c\ x\ t$ being the equivalent of an Otto cycle, and a–t–l–m being an additional amount of work realized from the same quantity of heat. The Sargent cycle is therefore somewhat more efficient

than is the Otto cycle, since it expands the gas more completely, reducing its terminal temperature (i.e., the temperature at point l) and therefore the amount of heat rejected.

The computations of the pressures, volumes, etc., of the Sargent cycle will be as follows: Let P_a, V_a, and T_a be respectively the pressure in pounds per square foot, the volume in cubic feet, and the absolute temperature of the charge at point a, and designate the same quantities for the charge at points c, t, x, l, m, and n by appropriate subscripts. Let H_a= the heat added at explosion and H_r the heat rejected at exhaust. We will then have for the swept volume of the cycle $V_m - V_c$. The weight of the charge will be

$$W = \frac{P_a V_a}{R T_a}. \quad \quad (1)$$

The volume of the compression space will be

$$V_c = V_a \left(\frac{P_a}{P_c}\right)^{\frac{1}{\gamma}}. \quad \quad (2)$$

The temperature of compression will be

$$T_c = T_a \left(\frac{P_c}{P_a}\right)^{\frac{\gamma-1}{\gamma}} = T_a \left(\frac{V_a}{V_c}\right)^{\gamma-1}. \quad \quad (3)$$

The rise in temperature resulting from the explosion will be

$$T_x - T_c = \frac{H_a}{W C_v}. \quad \quad (4)$$

The heat added as a result of the explosion will be

$$H_a = W C_v (T_x - T_c). \quad \quad (5)$$

The terminal temperature will be

$$T_l = T_x \left(\frac{V_c}{V_l}\right)^{\gamma-1} = T_x \left(\frac{P_l}{P_x}\right)^{\frac{\gamma-1}{\gamma}}. \quad \quad (6)$$

The fall in temperature at the end of expansion will be

$$T_l - T_a = \frac{H_r}{W C_v}. \quad \quad (7)$$

The heat rejected will be

$$H_r = W C_v (T_l - T_a). \quad \quad (8)$$

The efficiency of the cycle will be

$$E = \frac{H_a - H_r}{H_a} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

or

$$E = \frac{C_v\,(T_x - T_c) - C_v\,(T_l - T_a)}{C_v\,(T_x - T_c)}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

or

$$E = \frac{T_x - T_c - T_l + T_a}{T_x - T_c}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

It will appear from this equation that the efficiency of the Sargent cycle, unlike that of the Otto cycle, depends on the temperature of explosion. The work represented by the area, $c'\ c,\ a\ a'$, is

$$U_{ca} = \frac{P_c\,V_c - P_a\,V_a}{\gamma - 1} = R\left(\frac{T_c - T_a}{\gamma - 1}\right), \quad . \quad . \quad . \quad . \quad . \quad (12)$$

the work of expansion is

$$U_{xt} = \frac{P_x\,V_x - P_l\,V_l}{\gamma - 1} = R\left(\frac{T_x - T_l}{\gamma - 1}\right), \quad . \quad . \quad . \quad . \quad (13)$$

the work represented by the area $a'\ a.\ \ mm'$ is of course,

$$U_{am} = P_a\,(V_m - V_a); \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

the total work of the cycle is

$$\frac{P_x\,V_x - P_l\,V_l - P_c\,V_c + P_a\,V_a - P_a\,V_m + P_a\,V_c}{\gamma - 1}$$

$$= \frac{V_c\,(P_x - P_c) - V_l\,(P_l + P_a) + 2P_a\,V_a}{\gamma - 1}. \quad . \quad . \quad (15)$$

288. The Diesel Cycle Engine. Another cycle which had been employed for the internal combustion engine, and which is especially adapted for oil fuel, is known as the Diesel cycle. This also is a four-stroke cycle, and the card is shown in Fig. 160. During the first or exhaust stroke, the products of combustion are expelled from the cylinder. During the second or suction stroke, a charge of pure air is drawn into the cylinder. During the third or compression stroke, this air is compressed from point a to point c, the pressure rising to from 500 to 700 pounds per square inch.

At the end of the compression stroke, as the piston again moves forward, expanding the charge, a quantity of fuel is injected into the cylinder by means of a pump. As a result of the adiabatic compression, the temperature of the air has been raised to such a value that the fuel is kindled and burns as it flows into the cylinder, imparting heat to the charge. It was originally proposed that this fuel should be injected at such a rate that the expansion of the charge would be isothermal, the heat supplied by the burning fuel being just equal to the work performed by the expanding charge. After the injection of the fuel ceases, the charge expands adiabatically. The isothermal expansion line is the line $c\ d'$, and the

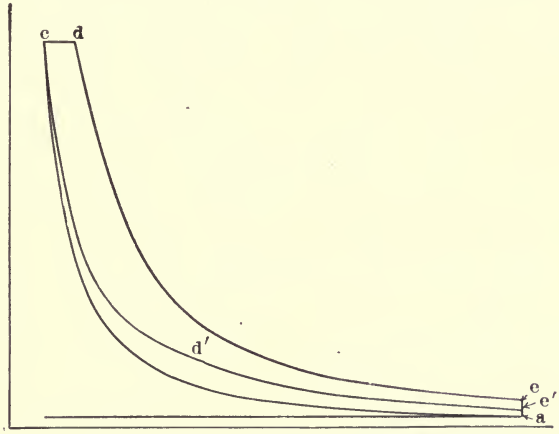


Fig. 160.—Theoretical cards from a Diesel cycle engine.

adiabatic expansion line, the line $d'\ e'$ in Fig. 160. Governing is effected by increasing or decreasing the quantity of fuel injected, which involves a change in the length of the isothermal, and in the final state of the working fluid at point e'. Increasing the quantity of fuel injected raises the final temperature of the charge and so reduces the efficiency of the cycle.

Isothermal expansion of the charge during the fuel injection period results in a cycle of high thermodynamic efficiency for a given temperature range. On the other hand, it is impossible to design a mechanism which will inject the fuel at exactly the proper rate to give isothermal expansion, and the mechanical efficiency of the engine employing it will be small. It has been shown in practice that it is better to admit the fuel at such a rate as to maintain the pressure practically constant during the early part of the working stroke. After all the fuel has been injected, adiabatic expansion commences.

The form of card given by such an engine is shown in the same figure. Line $c\ d$, representing the isobaric, and line $d\ e$ the adiabatic portion of the

expansion period. Comparing this card with the first one, it will be seen that the ratio of the mean to the maximum pressure is much greater in the case of isobaric than in the case of isothermal expansion, and that the mechanical efficiency of the engine is correspondingly better. In addition, the engine employing isobaric expansion gives much more power, although the parts of the two engines must be of the same weight and cost. The practical advantages will thus be seen to lie entirely with the cycle which employs isobaric expansion.

The methods of computing the states of the working fluid at the various points in the cycle are identical with those employed for the Otto cycle. The quantities of heat added or rejected, and the work performed during the several periods of the cycle may be readily computed. With a compression pressure of 600 pounds per square inch absolute, an original

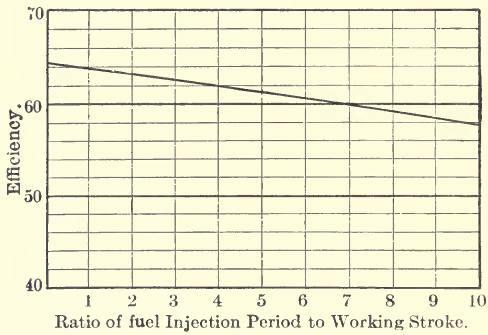


FIG. 161.—Relation between efficiency and fuel injection period for a Diesel cycle engine employing isobaric expansion.

pressure of 14.7 pounds per square inch, an original volume of 1 cubic foot, and an original temperature of 530° absolute, the compression volume will be 0.0714 cubic feet, and the compression temperature 1550° absolute, or 1090° F. The efficiency of the cycle under varying conditions of load may be seen by reference to Fig. 161, in which the abscissæ are swept volumes up to the point d, expressed as a per cent of the total swept volume $\left(\text{i.e., values of } \frac{100\,(V_b - V_c)}{(V_a - V_c)}\right)$ and ordinates are efficiencies.

PROBLEMS

1. An Otto gas engine has a compression pressure of 90 lbs. gage. Find the temperature of compression, assuming the air temperature to be 70° F.

Ans. 935° abs.

2. Find the efficiency of the cycle. Ans. 43.3%.

3. Find the explosion pressure, assuming the rise in temperature to be 2500°.

Ans. 386 lbs.-abs.

4. Find the terminal temperature. Ans. 1950° abs.

5. Find the work done per pound of working fluid. Ans. 142,300 ft.-lbs.

6. Find the volume of the compression space per pound of working fluid.
Ans. 3.31 cu.ft.

7. Find the terminal volume per pound of working fluid. Ans. 13.34 cu.ft.

8. Find the swept volume per pound of working fluid. Ans. 10.03 cu.ft.

9. Find the work done per cubic foot of swept volume per cycle per pound of working fluid. Ans. 14,200 ft.-lbs.

10. Assuming that a Sargent cycle engine, having the same compression as the Otto cycle engine in Problem 1 and the same explosion pressure as in Problem 3, expands its charge to atmospheric pressure, what is the terminal volume per pound of working fluid? Ans. 33.6 cu.ft.

11. Find the work of expansion. Ans. 273,000 ft.-lbs.

12. Find the work of compression. Ans. 52,700 ft.-lbs.

13. Find the work represented by the area $a'\ a\ m\ m'$ in Fig. 159.
Ans. 43,000 ft.-lbs.

14. Find the net work of the cycle per pound of working fluid.
Ans. 177,300 ft.-lbs.

15. Find the heat added per pound of working fluid. Ans. 423 B.T.U.

16. Find the efficiency of the cycle. Ans. 54.0%

17. In a Diesel cycle the compression is carried to 700 lbs. absolute. The initial temperature is 70° F. Find the compression temperature. Ans. 1620° abs.

18. Find the volume of compression space per pound of working fluid.
Ans. 0.86 cu.ft.

19. Assuming that the temperature of the working fluid is doubled during isobaric expansion, find the work done per pound of working fluid during isobaric expansion. Ans. 86,600 ft.-lbs.

20. Find the work done per pound of working fluid during compression.
Ans. 142,000 ft.-lbs.

21. Find the work done per pound of working fluid during adiabatic expansion.
Ans. 239,000 ft.-lbs.

22. Find the net work of the cycle per pound of working fluid.
Ans. 186,000 ft.-lbs.

23. Find the heat added. Ans. 384 B.T.U.

24. Find the efficiency of the cycle. Ans. 61.5%.

CHAPTER XX

NOTES ON THE DESIGN AND PERFORMANCE OF INTERNAL COMBUSTION ENGINES

289. Thermal Behavior of the Charge during Induction and Compression. In the theory of the Otto cycle developed in the previous chapter the charge entering the cylinder was assumed to have the properties and temperature of atmospheric air. Actually the charge consists of a mixture of air and combustible gas, and its properties are therefore somewhat different from those of pure air. Since the wall of the cylinder has a temperature somewhat higher than that of the atmosphere, the charge is heated as it enters the cylinder. As it is drawn into the cylinder, it mixes with a considerable volume of spent charge whose temperature is relatively high. In consequence, the temperature of the working fluid contained in the cylinder at the beginning of compression is almost always considerably higher than that of the atmosphere, and the effect is, of course, to reduce the power of the engine by reducing the weight of the working fluid.

During the compression stroke the temperature of the charge rises. As soon as its temperature exceeds that of the cylinder wall, it parts with heat by conduction and radiation. The compression is therefore not adiabatic, but is approximately polytropic, the actual compression line lying between the adiabatic and the isothermal lines. The index of the compression line ranges from 1.25 to 1.35, its exact value depending upon the size, the speed, and the design of the engine. In general the index of the compression curve will have a high value in the case of a large fast-running engine with a small area of wall surface exposed to the charge, and will have a low value when the opposite conditions obtain. Leakage has the same apparent effect upon the form of the compression curve as does heat absorption by the cylinder wall, and, like it, produces more serious effects in small or slow-speed than in large or high-speed engines. Leakage, however, is more detrimental than heat absorption in its effects upon the power and efficiency of the engine.

290. Thermal Behavior of the Charge during Ignition and Expansion. At the end of the compression stroke, combustion begins. In developing the theory of the Otto cycle, it was assumed that combustion took place instantly. Such is not the case in practice, since combustion is a

chemical reaction, and a chemical reaction is a progressive and not an instantaneous process. The rate at which a reaction occurs is variable, depending upon the temperature and density of the reacting substances, and upon the extent to which they are diluted by inert substances and the compounds produced by the reaction. The result of the combustion is to greatly increase the temperature of the charge, to reduce the density of the reacting substances, and to dilute them with the products of combustion. Each of these effects reduces the rate at which the reaction progresses, so that although it is rapid at first, the rate quickly drops off, and the reaction in theory takes an infinite time for its completion. When the temperature of the exploding charge reaches a value of about 3300° to 3600° absolute, which it does almost immediately, the reaction

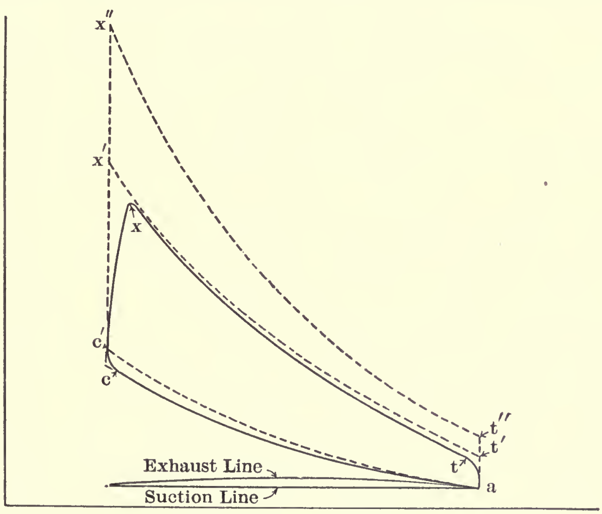


FIG. 162.—Actual and theoretical Otto cycle cards compared.

practically ceases, and further combustion takes place only after the temperature begins to fall. This phenomenon is known as **delayed combustion,** as **suppressed combustion,** and as **dissociation.**

As a result of the comparatively gradual and incomplete combustion of the charge, the rise in pressure resulting from the explosion takes an appreciable time, which modifies the card by giving it the form shown in Fig. 162, where the explosion line c–x is curved instead of being vertical, as would be the case were the combustion instantaneous. Furthermore the temperature and pressure realized as a result of the explosion are only a fraction of what they would be were the combustion instant and complete. Hence in order to obtain a given rise in pressure, and therefore a given quantity of work from the charge, it is necessary to use a larger quantity of fuel than theory would indicate. The actual efficiency

of the engine is therefore seriously reduced by the phenomenon of suppressed combustion. The most of the fuel remaining unburned at point x begins to burn as soon as the adiabatic expansion of the charge permits of a sufficient fall in temperature. This is known as after burning. The combustion is not absolutely complete at the end of the expansion stroke, but it is usually so far completed that only slight traces of unburned gases can be discovered in the exhaust.

The temperature of the charge, as a result of the explosion, usually reaches a value of from 3000° to 3600° absolute. At these extreme temperatures gases readily part with heat to their surroundings by conduction and radiation. The walls of the cylinder are of course cooled by a water jacket or other suitable means. Since the jacket maintains the walls at a temperature usually ranging from 150° to 250°, the rate at which the gases give up heat to the walls is very great. That portion of the charge which is in immediate contact with the wall is cooled by conduction almost to the temperature of the wall itself. The remainder of the charge is at a very much higher temperature and radiates its heat to the wall. As a result of the phenomenon of after burning the charge receives heat during the working stroke. At the same time, it is losing heat by conduction and radiation to the cylinder walls and by transforming it into the work of expansion. The expansion which takes place during the working stroke is therefore not adiabatic, and the rate of heat transfer is usually such that more heat is lost to the wall than is gained from the after burning of the charge. The index of the expansion line is therefore usually somewhat greater than the index of the adiabatic expansion line, sometimes ranging as high as 1.7, although its usual value lies between 1.41 and 1.45. The same operating conditions which tend to reduce the heat loss to the cylinder wall during the compression stroke tend to reduce the heat loss during the working stroke, but whereas these conditions tend to raise the value of the index of the compression curve, they tend to lower the value of the index of the expansion curve.

291. Form of the Actual Card. The form of the actual card obtained from an Otto cycle engine is that shown in full lines in Fig. 162. The line which would be obtained were the compression of the charge adiabatic is the dotted line ac'. It will be noted that the actual compression pressure is less than that which would result from adiabatic compression. In consequence, the actual compression temperature will also be considerably less than that resulting from adiabatic compression, and the efficiency of the actual cycle will therefore be less than the efficiency of theoretical cycle.

Were the combustion of the charge instantaneous, the line cx would be vertical. Since such is not the case, the line will have the general form shown; its exact form depending on the instant at which ignition

commences, the speed and size of the engine, the character of the charge, etc.

The actual expansion line x–t, falls below the adiabatic expansion line, $x't'$, since its index is greater than the index of the adiabatic line. Were the entire heat of combustion contained in the charge utilized in instantly raising its temperature, and were the specific heat of the charge constant, its pressure would rise to the point x'' as a result of the explosion, the adiabatic expansion line would be the line $x''\ t''$, and the work of the cycle would be very greatly increased.

On account of wire drawing and fluid friction, the corner of the card at t is rounded, and the suction line falls below the exhaust line in the manner already shown in Fig. 154. This results in reducing the pressure at the beginning of compression, and the loop enclosed between the suction and exhaust lines represents negative work.

292. Losses in the Gas Engine. The losses which occur in an Otto cycle engine may be classified as follows:

First: Losses due to delayed or suppressed combustion.

Second: Losses due to the radiation and conduction of heat to the cylinder walls.

Third: Losses due to the leakage of the charge.

Fourth: Exhaust losses or losses due to the sensible heat of the charge at the termination of expansion.

Fifth: Losses due to fluid friction and wire drawing of the charge.

Sixth: Losses due to the mechanical friction of the engine.

It is obvious that the efficiency of the actual cycle must always be less than that of the theoretical cycle. Hence the sum of the six losses is always greater than the exhaust loss of the theoretical Otto cycle having the same compression pressure, and supplied with the same quantity of heat energy. The theoretical loss made necessary by the form of the cycle can be reduced only by increasing the compression pressure. As the compression pressure is increased, the explosion pressure is increased in very nearly the same proportion. This makes necessary a corresponding increase in the strength and weight in the parts of the engine, and increases the cost of its construction. Compression pressures higher than 200 pounds gage are therefore not practicable except when very lean fuels are employed. When fuels of high heating value (such, for instance, as gasoline vapor), are employed, the compression pressure must be much less than 200 pounds in order to avoid excessive explosion pressures.

The effect of delayed combustion is, of course, to reduce the explosion pressure and therefore to reduce the amount of work performed during the cycle and the efficiency of the engine. Were the charge to reach the temperature which would result from complete combustion, the loss to the cylinder walls would be greater than is actually the case. Since the

combustion of the charge continues during the working stroke, a small part of the heat so developed is transformed into work, but the most of it is rejected in the sensible heat of the exhaust. Any portion of the charge remaining unburned at the instant of release is, of course, rejected and the potential heat contained in it is lost. The effect of suppressed combustion is therefore to increase the loss in the exhaust and to decrease the loss to the water jacket. The loss from suppressed combustion may be diminished by the employment of a lean charge (i.e., one which has a comparatively low heating value). The effect of the lean charge is to reduce the theoretical maximum temperature of explosion, and therefore to make possible a quicker and a more complete burning of the charge.

The effect of the heat transfer from the charge to the walls of the cylinder, where it is absorbed by the jacket water, is to reduce the work required for compression, the explosion pressure, and the work done during the expansion stroke. The net result is that the work done during the cycle is diminished somewhat, and the efficiency of the cycle is unfavorably affected. A considerable part of the heat transferred to the water jacket is, however, heat that would otherwise be rejected at exhaust, and the amount of heat lost to the water jacket is not a true measure of the power and efficiency lost as a result of water jacketing. While it is well to reduce the jacket loss to minimum, it is not as serious in its effects upon the engine efficiency as are other forms of losses.

The effect of leakage during the compression stroke is to allow a portion of the charge to escape after work has been done upon it, but before it has returned any portion of this work. Furthermore it carries away the potential heat of combustion, which is, of course, entirely lost. Leakage during the expansion stroke does not affect the efficiency of the engine so seriously, but it does produce some loss by lowering the mean pressure during the expansion period. It might be thought that on account of the high pressures employed, leakage would be a serious matter in the case of the gas engine. When, however, the valves, the piston, the cylinder, and the rings are in good order, no appreciable leakage can take place.

The exhaust loss may be reduced by expanding the charge more completely, as is done in the Sargent cycle engine. The additional amount of work obtainable in this way is not, however, very great, as may be seen by reference to Fig. 159, in which the area *a t l m* represents the power usually obtained in such a cycle from the exhaust losses of the Otto cycle. It will be seen that the area in question is a rather small portion of the entire area of the card. It has often been proposed to save some of the exhaust loss by compounding the gas engine (i.e., by discharging the exhaust from the first cylinder into a larger cylinder in which it may be more completely expanded). It will usually be found, however,

that the amount of work realized after deducting the extra losses incurred in transferring the charge from one cylinder to another, will not be sufficient to overcome the friction of the added cylinder. The employment of the Sargent cycle is a preferable alternative for utilizing the available energy of the exhaust.

The losses due to friction and wire drawing of the charge may be reduced by the employment of large valves which are opened promptly by properly designed mechanism. These losses increase with the speed of the engine, but do not become serious at the speeds ordinarily employed in stationary practice.

The conditions of high mechanical efficiency in the gas engine are the same as in the steam engine. It is not possible, however, to improve the mechanical efficiency of the gas engine by compounding, as may be done in the case of the steam engine. The mechanical efficiency of the gas engine may be improved only by careful attention to the details of the design of the mechanism and to the lubricating system. The friction losses are usually from 50 to 100 per cent higher in gas engines than in steam engines of equal power.

It will be seen from the above discussion that the conditions which favorably affect the efficiency of the gas engine are, in general, a high speed of rotation, the use of units of large power, the adoption of that form of cylinder which reduces the wall area per pound of charge per cycle to a minimum, and the employment of a lean charge. The most serious loss is that due to the delayed combustion of the charge. The Diesel cycle engine avoids the difficulty of delayed combustion and therefore gives promise of higher practical efficiency than does the Otto cycle engine, although other forms of loss (e.g., loss due to leakage) produce more serious results in the Diesel engine than in the Otto engine.

293. Limits of the Rotational Speed of Internal Combustion Engines. It has already been pointed out that the higher the speed at which a gas engine operates, the greater will be its efficiency. At ordinary speeds the power of the engine is increased by increasing the speed, since at ordinary speeds the work per cycle remains practically constant, and the number of cycles increases in direct proportion to the speed. It is not possible, however, to indefinitely increase the power of an internal combustion engine by increasing its speed. As the speed of the engine increases, the effect of wire drawing in reducing the quantity of charge taken in per cycle also increases. At speeds below 400 or 500 revolutions per minute, this effect is scarcely noticeable when the engine is equipped with mechanically operated valves. At speeds greater than this, however, the net work per cycle gradually falls off on account of the wire drawing of the charge. The speed at which the power of the engine reaches its maximum value depends upon the size and form of

the valves and ports. In general the larger and straighter the gas passages, the higher will be the speed at which the maximum power of the engine is realized. With ports of the usual proportions, it will be found that small two-cycle engines deliver their maximum power at from 700 to 900 revolutions per minute, while four-cycle engines deliver their maximum power at from 1200 to 1800 revolutions per minute. At speeds higher than this, the quantity of charge taken per cycle by the engine diminishes at a faster rate than the speed increases and the power of the engine falls off. At very high speeds, ignition fails from lack of sufficient compression of the charge, and the speed of the engine finally reaches a maximum where the power developed is just equal to that absorbed by friction.

In the case of stationary engines, the speed is, of course, very much lower than in automobile and other light high-speed engines. The usual speed for very large engines (i.e., engines of over 1000 horse-power) is from 75 to 150 revolutions per minute, the present tendency being to increase these speeds. In the case of single-acting stationary engines, the speed is usually from 200 to 300 revolutions per minute, although higher speeds are possible. As a usual thing, the speed of a large gas engine is limited by the highest speed at which the valve motion will work quietly and without undue wear. It will thus be seen that the speed of a gas engine is really limited by the design of its parts, and will of necessity be lower in the case of a large engine having heavy parts than in the case of a small engine having light parts. There is no reason, however, why much higher speeds may not be employed in stationary service, with an accompanying gain in economy of operation.

294. The Design of Internal Combustion Engines. In designing an internal combustion engine, it is usually sufficient to assume that the charge is taken in at atmospheric pressure and temperature, that the compression and expansion lines are polytropic, that the index of the compression line is 1.35, that the index of the expansion line is 1.45, that the explosion occurs instantly, that the rise in temperature as a result of the explosion is 2500°, that the specific heats of the charge are those of pure air; that the card factor is about 90 per cent, and that the mechanical efficiency of the engine is 85 per cent. Where actual cards from engines of practically similar design and approximately the same speed are available, corrections may be made in these figures. The work of compression and of expansion per pound of working fluid may be computed by the methods outlined in Art. 283. Their difference is the net work per pound of working fluid. The volume per pound of working fluid at the beginning and end of compression is next computed. The difference between these two volumes is the swept volume per pound of working fluid. Dividing this into the net work per pound

of working fluid, we will have the net work per cubic foot of swept volume per cycle, a quantity which we may designate by the letter W. The indicated horse-power of the engine will then be

$$HP = \frac{0.9\, W\, V\, N}{33{,}000},$$

in which HP is the indicated horse-power of the engine, W is the net work per cubic foot of swept volume per cycle, N is the number of cycles (i.e., explosions) per minute, and V is the swept volume of the cylinder in cubic feet. The brake horse-power of the engine at maximum load will be about 85 per cent of its indicated horse-power at maximum load. A gas engine is usually rated at from $^2/_3$ to $^3/_4$ of the maximum brake horse-power which can be obtained under the most favorable conditions of operation.

After obtaining the cylinder dimensions and the form of card, the remainder of the design of a gas engine is a matter of proportioning the parts to properly resist the strains which come upon them, and to arrange the valve mechanism so that it operates with a minimum of shock and wear. The design of the details of a gas engine is very similar to the design of the same parts of a steam engine, the only difference being produced by the greater shocks and higher pressures encountered in gas engine work, and the necessity of thoroughly water-jacketing or otherwise cooling all parts exposed to the working fluid.

295. Methods of Ignition. Three methods have been employed for igniting the charge of a gas engine, namely, by an electric spark, by contact with hot metal, or by contact with a flame. The latter method, although formerly much used, is now completely out of date, while the second method, known as hot tube ignition, is seldom used except for stationary engines operating on natural gas. In the early types of gas engines in which little or no compression was used, a flame was kept burning in a separate chamber and by opening a slide valve in a passage connecting this chamber with the cylinder of the engine at the proper point in the stroke, the flame was communicated to the charge. So long as the compression pressure was low, and the service required of the engine was not severe, this method of ignition was fairly satisfactory. It was, however, soon superseded by the method known as hot tube ignition.

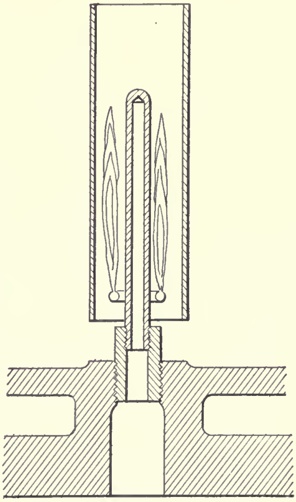


Fig. 163.—Hot-tube igniter.

The hot tube igniter, which is illustrated in Fig. 163, consists of a passage in the cylinder wall which terminates in a tube of wrought iron or nickel, kept heated by means of an argand flame which surrounds it. At the beginning of the compression stroke, the tube is filled with spent charge. As the compression proceeds, the spent

charge is compressed into the hot part of the tube and finally, near the end of the compression stroke, some of the fresh charge enters the hot part of the tube and is ignited. By its expansion, a flame is forced into the cylinder, and the main body of the charge thus ignited. It will be seen that the point in the cycle at which the explosion occurs depends upon the relative volume of the hot tube and the connecting passage, and upon the degree of compression. If ignition fails, or is too late, the volume of the hot tube must be increased. If ignition is early, the length and volume of the connecting passage must be increased, or the volume of the tube decreased. Other things being equal, the higher the degree of compression, the earlier the time of ignition. So long as the quality of the fuel supplied to the engine is uniform and the conditions of operation are steady, hot tube ignition is fairly satisfactory with comparatively high compression pressures. It can only be used, however, with hit-and-miss governing.

Electric ignition is effected in either of two ways, the first being known as the "**jump spark**" method and the second as the "**make-and-break spark**" method. The principle of the jump spark is illustrated in the diagram shown in Fig. 164. In this

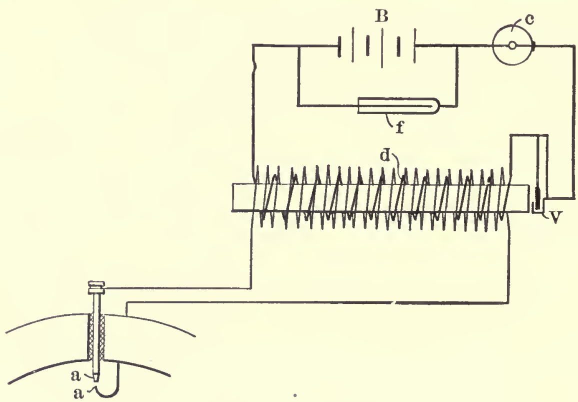


FIG. 164.—Diagram of a jump spark ignition apparatus.

diagram *a–a* are two platinum terminals contained within the cylinder of the engine in contact with the charge and separated by about $^1/_{32}$ of an inch. These terminals are connected with the induction coil shown, which gives a high voltage. At the proper time in the revolution of the engine, a connection is made in the primary circuit of this coil by means of a **commutator** or contact maker operated by the engine. The flow of primary current through the coil produces a secondary current of immensely higher voltage and much lower amperage, which passes between the terminals of the spark plug, igniting the charge. *B* is a battery or other source of current for the primary circuit of the induction coil. This current passes through the commutator *c* when it is in the position shown, thence through the vibrator *V*, through the primary circuit *d* and back to the battery. The vibrator is a device for interrupting the current and causes the primary current to be pulsating in character. The secondary current, which is induced by the presence of the pulasting primary current, is similar in character, but since the primary circuit of the induction coil consists of only a few turns, while the secondary circuit consists of many hundreds of turns of wire, all wound around a soft iron core, the voltage of the secondary current will be sufficiently great to force it to jump the terminals of the spark plug. These

terminals must of course be carefully insulated from one another or the current will be short-circuited instead of passing through the charge which is to be inflamed. A condenser f is usually connected into the primary circuit of the induction coil in order to increase the effectiveness of the apparatus.

The apparatus used for the make-and-break spark is much simpler, although it has the disadvantage of employing a movable part within the cylinder. The current originates in a battery B as shown in Fig. 165, passes through the spark coil C, which is a coil of wire surrounding a soft iron core, through the platinum terminals of the make-and-break spark plug P, and then returns to the battery. At the instant when it is desired to inflame the charge, the terminals being in contact, they are quickly separated and an arc is created which effects ignition. The purpose of the spark coil C is to intensify the arc by its inductive action. In order to economize current

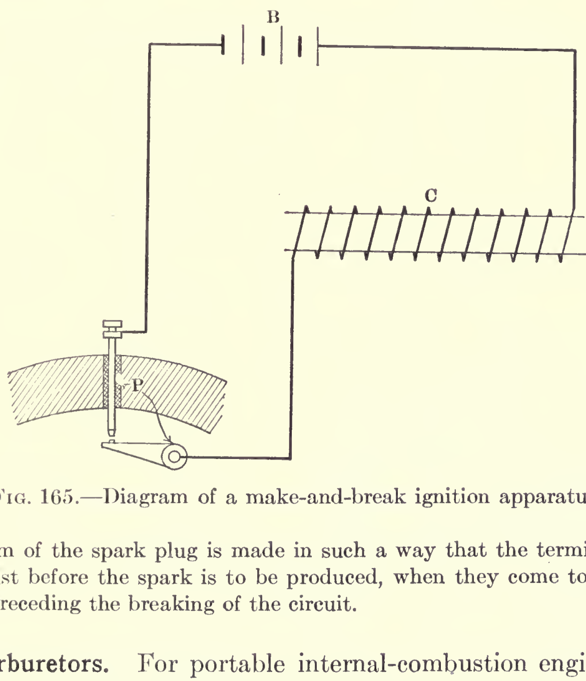


Fig. 165.—Diagram of a make-and-break ignition apparatus.

the mechanism of the spark plug is made in such a way that the terminals are separated until just before the spark is to be produced, when they come together for the instant just preceding the breaking of the circuit.

296. Carburetors. For portable internal-combustion engines such as are used in automobiles, launches, etc., it is customary to use as the fuel a hydro-carbon vapor usually obtained by the evaporation of gasoline. The gasoline is evaporated and mixed with the air which forms the remainder of the charge, in an apparatus known as a **carburetor.** Gasoline, when sprayed into air, rapidly evaporates at all ordinary temperatures, and fills the air with its vapor. If the air is allowed to become saturated with the gasoline vapor, the quantity of vapor contained in the air will be so great that the mixture is not explosive. A charge in which too little gasoline vapor is mixed with the air, will also fail to ignite. The office of the carburetor is then to introduce into the current of air entering the cylinder of the engine, a proper quantity of gasoline,

in such a manner that it will be completely evaporated and thoroughly mixed with the air.

The simplest form of carburetor is illustrated in principle in Fig. 166. It consists of a bowl or reservoir in which the gasoline is maintained at a constant level by means of a **float feed-valve.** This valve consists of a ring *r*, usually of cork, attached to a lever and pivoted in such a way that when the level of the gasoline sinks, the weight of the float will open the feed valve *f*, admitting more gasoline. From this chamber the gasoline flows to a small nozzle *n*, through a valve termed the **needle valve.** By varying the opening of this needle valve, the quantity of gasoline delivered through it in a given time by a given head of gasoline, may be varied. The needle valve is placed in a restricted passage in the air inlet. During the suction stroke of the engine, a quantity of air is drawn through this restricted passage at high velocity, and in consequence, the pressure of the air in the passage is less than the pressure of the atmosphere. The difference in pressure causes a jet of gasoline to flow from the needle valve in the form of a fine spray, and to mix with the on-rushing current of air. A second supply of air is taken through the check valve *v*, termed the auxiliary inlet, at a point beyond the needle valve, and mixes with the air containing the gasoline vapor. By adjusting the level of the gasoline in the float-feed chamber, the opening of the needle valve, and the strength of the spring controlling the opening of the auxiliary air inlet, the quantity of gasoline vapor in the charge of air may be controlled and a correct mixture obtained at all ordinary engine speeds. Most forms of carburetors at present on the market are modifications of the apparatus described. The arrangement of the several parts and the general appearance of the apparatus varies greatly. In some forms of the carburetor, however, the air is

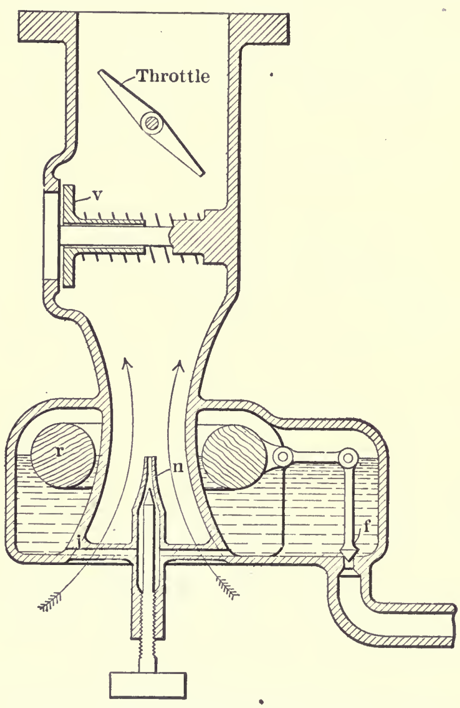


FIG. 166.—Simplified diagram of a carburetor.

drawn over a small quantity of gasoline contained in a bowl, instead of spraying the gasoline into the air. Such carburetors are not generally provided with auxiliary air inlets.

A float-feed carburetor is not usually used for furnishing gasoline vapor to stationary engines. In the case of stationary engines, the speed is usually constant and the range of adjustment required of the gasoline vaporizer is very much less. In some cases, the incoming change of air is drawn over a pan in which the gasoline is maintained at constant level. In other cases the gasoline is simply allowed to leak through a needle valve into the pipe through which the air supply is drawn. Neither method of vaporization is as satisfactory, however, as the use of a carburetor.

297. The Testing of Internal Combustion Engines. The following quantities must be determined in making a complete test of an internal combustion engine of the Otto type.

First, the pressure and temperature of the atmosphere and of the gas in case a gaseous fuel is employed.

Second, the heating value of the fuel.

Third, the weight or volume of the fuel supplied to the engine.

Fourth, the volume of the air supplied to the engine.

Fifth, the number of revolutions per minute.

Sixth, the number of cycles per minute.

Seventh, indicator cards are taken from the cylinders.

Eighth, the weight of jacket water used, and its initial and final temperature.

Ninth, the brake horse-power of the engine.

The precautions which must be taken in making such a test to insure that the data are properly taken, have been outlined by a committee of the A.S.M.E., and the rules have been published by the society in pamphlet form. It is important that the conditions throughout the test should be as nearly uniform as possible. Readings should be taken at frequent intervals, say every ten minutes.

A gas engine test may be analyzed graphically in a manner similar to that already described in Art. 195. The actual card of the engine is superimposed upon the theoretical card which would be given were the same quantity of heat added to a charge of pure air, as was actually introduced into the engine, per cycle, in the fuel used.

After determining the indicated horse-power shown by the different sets of cards taken during the test, that card should be chosen whose form and area are nearest to the average. The heat supplied per pound of charge is next computed, and from the dimensions of the engine and the pressure of the atmosphere a theoretical card is drawn. This card

is shown in dotted lines in Fig. 167. Upon this theoretical card, to the same scale of pressures and volumes, is superimposed the card shown from the test as best representing the average conditions. Usually the beginnings of the compression lines coincide on the two cards, since the effect of wire drawing in reducing the pressure of the charge at the beginning of compression is inappreciable. The compression line of the actual card *a–c* will, however, fall below that of the theoretical card *a–c′* in the manner shown.

If the charge were compressed to point c_1, and its combustion were then instant and complete, the heat added would raise the pressure to some point x'', whose position may be computed. If the charge then

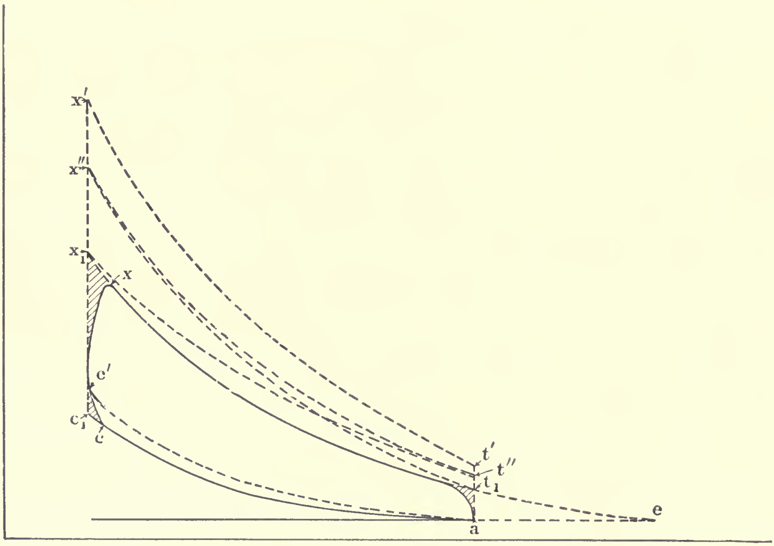


Fig. 167.—Graphical analysis of the losses in an Otto cycle engine.

expanded adiabatically following the line $x''t''$, the engine would give the card a–c_1–x''–t''. The difference between the area x'–t'–t''–x'' and the area a–c'–c_1 will then be the power lost on account of the heat transferred to the walls of the cylinder during the compression stroke. The area t_1–x_1–$x''t''$ then represents the work lost on account of the combined effects of suppressed combustion and heat loss to the cylinder wall during the working stroke. Were the combustion not suppressed, the expansion line would probably have approximately the form $x''t_1$. Were there after burning but no heat transfer to the cylinder wall, the expansion line would have approximately the form x_1t''. The available exhaust loss which might be recovered by complete expansion of the charge,

is represented by the area t–e–a, which is the power which would be obtained from the charge were it expanded adiabatically down to atmospheric pressure. The remainder of the exhaust loss cannot be recovered by expansion of the charge. The shaded areas represent the loss due to fluid friction, and to the fact that it takes an appreciable time for the explosion pressure to reach its maximum.

From a complete internal combustion engine test a heat balance may be made out showing the actual distribution of the heat supplied to the engine. The proportion of the heat lost in friction and that transformed into useful work may be readily computed from the brake and the indicated horse-power of the engine. The heat transferred to the jacket water during the test may be found by measuring the water used and obtaining its rise in temperature. The sensible heat rejected at the exhaust may be found by computing the temperatures of the charge at point t_1 and at point a and multiplying the difference by the specific heat of the charge at constant volume. The remainder of the heat supplied is rejected in the exhaust in the form of unburned combustible, or is radiated from the engine, or represents the error of the test. It may be noted that the temperature of the exhaust, as obtained by a thermometer, is not the temperature corresponding to the point T_1, but is lower on account of the work performed by the exhaust in expansion down to atmospheric pressure against the resistance of the air.

298. Actual Efficiencies of Internal Combustion Engines. The total efficiency of a good gas engine usually ranges between 20 and 30 per cent. Efficiencies as high as 38 per cent have been claimed, but it is very doubtful if total efficiencies higher than 32 per cent have ever been realized. In Table XIV will be found the results of typical tests of different forms of internal combustion engines. These are not the highest efficiencies which have been realized, but are those which have been realized continuously in service.

It will be seen that the efficiency of the internal combustion engine is higher than that of the steam engine or steam turbine. The cost of fuel is therefore smaller in the case of the internal combustion engine than in the case of a steam engine or steam turbine. The cost of attendance is also smaller. On account of the high cost of an internal combustion engine plant, however, the fixed charges are large. In units of small power, the cost of fuel and of attendance is the principal item of expense. In units of large power the fixed charges upon the investment become the principal item. In general, it will be found that in small powers, the internal combustion engine will be the cheapest one to operate, while in large powers the steam plant, and more especially the steam turbine plant, may be operated at the minimum of expense. When, however, the cost of fuel is high, as it is in certain parts of the

world, the internal combustion engine is the prime motor of the highest commercial efficiency.

TABLE XIV

EFFICIENCIES OF MODERN INTERNAL COMBUSTION ENGINES

Kind of Fuel.	Nominal Brake H.P.	Pounds Fuel per B.H.P. Hour.	Cubic Feet Fuel per B.H.P. Hour.	Heating Value of Fuel.	Producer Efficiency Per Cent.	B.T.U. per B.H.P. Hour	Mechanical Efficiency Per Cent.	Total Efficiency Per Cent.
Alcohol	14.0	1.00	—	10,440	—	10,440	—	24.4
Illuminating gas	50.0	—	17.75	560	—	9,950	86	25.6
Suction producer gas from anthracite	20.0	0.995	—	14,940	86.5	12,800	—	19.9
Kerosene (Diesel cycle)	30.0	0.483	—	18,550	—	9,000	72	28.3
Blast-furnace gas	1200	—	—	—	—	9,080	83.1	28.1
Coke oven gas	620	—	—	—	—	9,400	70.3	27.1
Producer gas	483	0.965	—	14,321	73.8	10,200	83.8	25.0

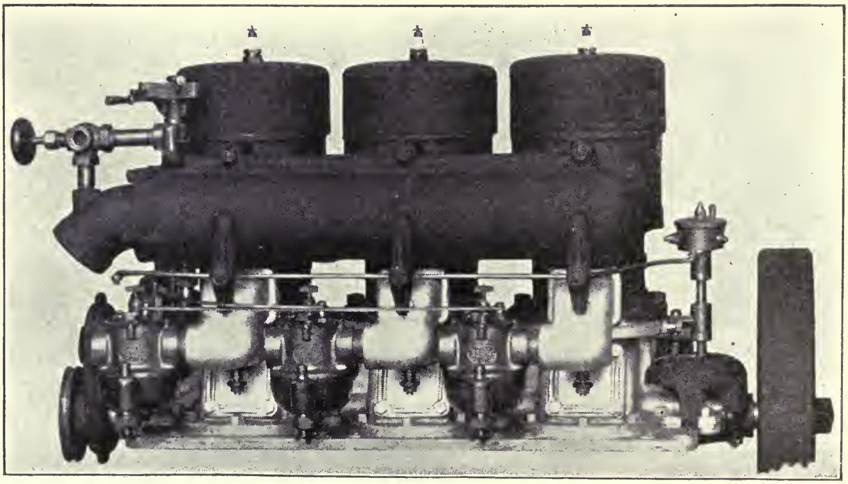


FIG. 168.—Three-cylinder two-cycle marine engine.

In Fig. 168 will be found an illustration of a small two-cycle engine, using gasoline for fuel, of a type often employed for propelling small boats. This is a three-cylinder engine with three separate carburetors and jump spark ignition, and is governed by throttling the charge and delaying the spark.

In Fig. 169 will be found an illustration of a medium sized (50 horse-power) four-cycle stationary engine, adapted for operation with illuminating and natural gas. Engines of this type are also frequently provided

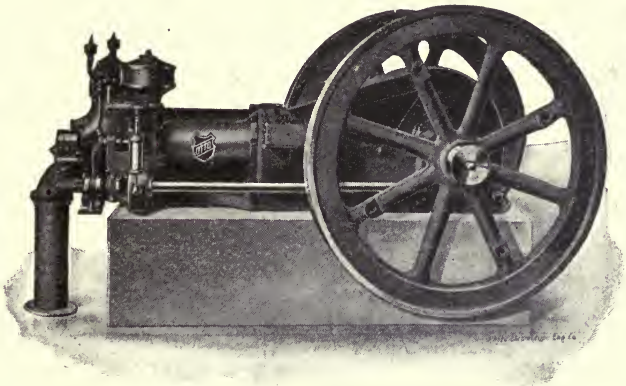


Fig. 169.—Four-cycle stationary gas engine.

with suction producer plants, and operate on producer gas generated from anthracite coal.

In Fig 170 may be seen a view of part of an installation of 17 gas engines, direct connected to 2000 kilowatt alternating current generators. This installation is at the Geary plant of the United States Steel Company and the engines were built by the Allis-Chalmers Company. The engines are of the four-cycle type, operate on blast furnace gas, and have four double acting cylinders each. By the use of four double acting cylinders in this manner, the engine is made to work with as much smoothness as does a cross compound steam engine. Gas engines of this type have been built of 5400 horse-power.

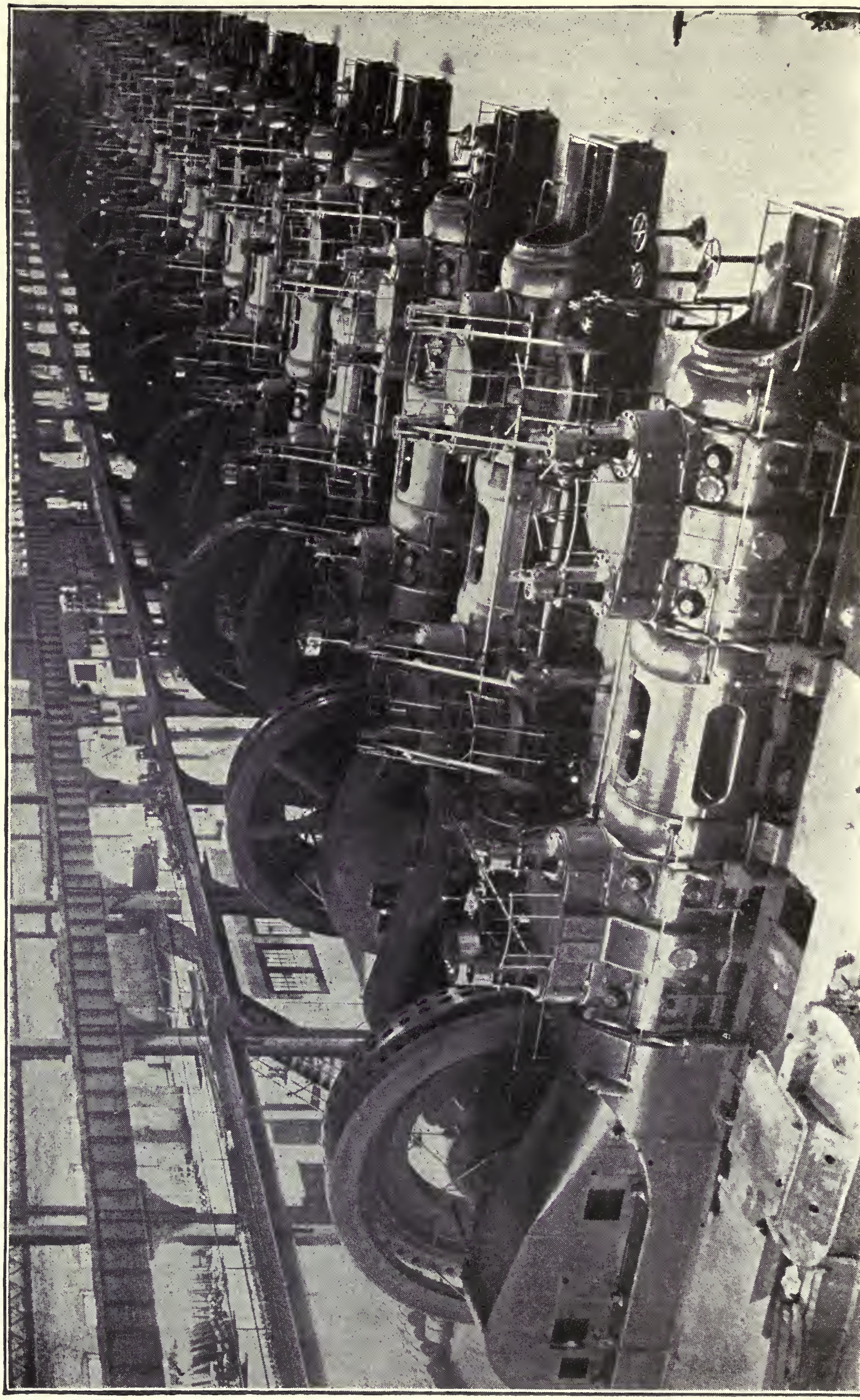


Fig. 170.—Part of an installation of seventeen 2000 K.W. four-cylinder four-cycle double-acting engines operating on blast furnace gas.

PROBLEMS

1. An Otto cycle engine having a compression pressure of 150 lbs. gage is to be designed. Assume atmospheric pressure to be 14 lbs. per square inch absolute and atmospheric temperature to be 70° F. Find the initial volume per lb. of working fluid. Ans. 14.03 cu.ft.

2. Find the volume of the compression space per pound of working fluid. Ans. 2.43 cu.ft.

3. Find the work of compression. Ans. 68,900 ft.-lbs.

4. Find the temperature resulting from the compression. Ans. 978° abs.

5. Find the explosion pressure. Ans. 533 lbs. per sq. in.

6. Find the terminal pressure. Ans. 42 lbs. per sq. in.

7. Find the work of the expansion. Ans. 226,000 ft.-lbs.

8. Find the net work done per pound of working fluid. Ans. 157,100 ft.-lbs.

9. Find the net work done per cubic foot of swept volume. Ans. 13,550 ft.-lbs.

10. The engine is required to have a nominal brake horse-power of 50. What will be the required indicated horse-power if the nominal brake horse-power is assumed to be $^3/_4$ of the maximum brake horse-power? Ans. 78.5 H.P.

11. If the engine makes 75 cycles per minute, find the number of cubic feet of swept volume per cycle required in order to develop this indicated horse-power. Ans. 2.55 cu.ft.

12. What cylinder diameter will be required if the piston speed is 600 ft. per minute, and the engine is single acting? Ans. $15\frac{1}{4}$ ins.

CHAPTER XXI

GASEOUS FUELS

299. Classification of Fuel Gases. The gases usually employed as fuels may be divided into six classes. They are:

1. Coal gas.
2. Coke-oven gas.
3. Water gas.
4. Natural gas.
5. Producer gas.
6. Blast-furnace gas.

Gases of the first three classes are usually termed **illuminating gas**, since they were originally made and sold for lighting purposes.

300. Coal Gas. Coal gas, often termed **bench gas** in order to distinguish it from by-product coke-oven gas, is obtained by the destructive distillation of bituminous coal, being formed from the volatile matter of the coal. In the bench process the coal is heated by means of a coke fire in small cast-iron or clay retorts, holding two or three hundred pounds of coal each. The volatile matter expelled from the coal consists largely of hydrocarbon vapors the greater part of which are decomposed by the heat into carbon and permanent gases. The substances evolved from the coal are removed from the retorts by means of a pump termed an exhauster. They consist of water vapor, ammonia, condensible hydrocarbons and fixed gases. By cooling the products of the distillation in a suitable apparatus, the water and the condensible hydrocarbons are removed. The gas is then passed through a scrubber (i.e., a tower filled with wooden checker work, coke or some similar material through which water trickles). The water in the scrubber absorbs the ammonia and removes the dust and the final traces of the condensible vapors. The gases are then passed through purifiers, which are chambers containing trays of sesquioxide of iron, or of lime, which remove the sulphur compounds from the gases. The gas remaining is a mixture of permanent gases consisting usually of from 38 to 48 per cent of hydrogen, 2 to 14 per cent of carbon dioxide, 31 to 43 per cent of marsh-gas, $4\frac{1}{2}$ to $7\frac{1}{2}$ per cent of olefiant gas and 1 to 3 per cent of nitrogen. The heating value of the gas usually ranges from 550 to 650 B.T.U. per cubic foot. The products of combustion are, of course, carbon dioxide and water vapor. The

density of the gas is variable, usually ranging from 25 to 50 per cent of that of air. Coal gas is stored over water in large inverted tanks called gasometers, and distributed to consumers by means of pipe lines. The cost of bench gas to small consumers usually ranges from $0.80 to $1.50 per thousand cubic feet, and it is occasionally sold as cheaply as 40 cents per thousand cubic feet to large consumers.

The by-products of the process as well as the gas itself have a commercial value. The tar recovered from the condensing apparatus is a source of several thousand chemicals of great commercial value, the ammonia recovered is of value in chemical industries, and that portion of the coke not used for heating the retorts is of value as a fuel. However, the commercial efficiency of the bench process of gas-making is very low, and it is not profitable to use such a gas as a fuel except when comparatively small quantities of fuel are wanted.

301. By-product Oven Gas. Coke is in great demand in the metallurgical industries. Formerly it was prepared in a type of kiln usually termed a **bee-hive coke oven,** in which all of the by-products resulting from its manufacture were wasted. It is now often made in a form of oven termed a **by-product coke oven,** in which the by-products resulting from its manufacture are saved and utilized. The by-product coke oven consists of several retorts of firebrick placed side by side in a battery. The retorts are of sufficient size to contain from 6 to 8 tons of coal each. The walls of the retorts enclose flues which are heated by a gas flame. The charging and discharging of the retorts are effected by machinery. Regenerators are provided for preheating the air and gas, used in heating the retorts, so that no heat is wasted by the escape of hot gases or products of combustion. In consequence of the heating of the retort the coal contained in it is coked and the volatile matter driven off. The gas which first comes from the oven is rich in hydrocarbons and is therefore suitable for illuminating purposes. The gas which comes from the oven after the coking process is nearly completed is deficient in illuminants on account of the high temperature at which it is distilled. It is therefore suitable only for fuel gas, and a portion of it is employed to heat the retorts. The illuminating gas which comes from the oven is similar in character to the gas obtained by the bench process, and is treated in a similar way in order to obtain from it the same by-products and the same quality of gas. The principal difference between the bench process and the by-product coke-oven process lies in the fact that the coal is handled in large quantities in the latter process, that the cost of labor is very much less and that the value of the coke produced is much greater on account of its superior quality. The gas from a by-product coke-oven plant may be stored in exactly the same way as gas from a bench-process plant and is suitable for exactly the same purposes.

The cost of making gas by the by-product coke-oven process is much less than by making it by the bench process, however, and it may on that account be employed as a fuel when the cost of bench gas would be prohibitive. By-product coke-oven gas makes an ideal gas-engine fuel, and the process is especially suitable for use in connection with a metallurgical industry in which coke and power are both needed.

302. Water Gas. Water gas is made by passing a current of steam through a bed of incandescent coke in a suitable retort. The steam is decomposed to hydrogen and oxygen and the oxygen unites with the carbon to form carbon monoxide. The reaction is

$$\underset{12}{C} + \underset{18}{H_2O} = \underset{28}{CO} + \underset{2}{H_2}$$

It will be seen from this reaction that in theory water gas consists of equal volumes of carbon monoxide and hydrogen. By weight it consists of 6.67 per cent of hydrogen and 93.33 per cent of carbon monoxide. Its heating value per pound is

$$0.933 \times 4380 + 0.067 \times 62{,}000 = 8230 \text{ B.T.U.}$$

Its heating value per cubic foot will therefore be found to be 341 B.T.U. Such gas is not suitable for use as an illuminant in an open-flame burner, since it burns with a colorless flame. Hence, when it is intended to sell such gas for illuminating purposes, it is customary to add to it vapors obtained by the distillation of petroleum residue, a process termed **enriching.** These vapors are known as illuminants, and add to the heating as well as the illuminating value of the gas.

The decomposition of the steam and the consequent formation of hydrogen within the generator absorbs heat. As a result, the temperature of the bed of coke falls, and the reaction would soon cease if heat were not supplied in some manner. This is usually done by passing a current of air through the coke bed within the generator, by means of which a portion of the bed of coke is burned and the temperature of the whole mass is raised. When the temperature has reached a sufficiently high value, the current of air is stopped and a current of steam takes its place. As a result of the reaction, the temperature of the bed of coke is again reduced until finally the reaction ceases and the gas coming from the producer consists principally of steam, instead of combustible gases, when the current of air is again introduced in order to raise the temperature of the coke. In order to make the water-gas process continuous, it is therefore necessary to employ two or more generators, so that one may be warming up while the other is generating gas. By the employment of a suitable regenerator system, the sensible heat which would be carried

away in the current of air used to raise the temperature of the coke, may be saved, so that the entire heating value of the coke will finally appear in the heating value of the gas produced from it, except for such unavoidable losses as are due to radiation and to inefficiency of the regenerator system. It is not usual, however, to employ a regenerator system in small water-gas plants, so that the heating value of the gas generated from a given quantity of coke is usually only from 50 to 60 per cent of the heating value of the coke.

Water gas is usually much cheaper than coal gas, and is therefore, of more practical use as a fuel. Water gas, as usually made and sold for illuminating purposes, after enriching, has a density of 0.6 of that of air, and a heating value of about 600 B.T.U. per cubic foot. Besides the illuminants, it consists principally of carbon monoxide and hydrogen, together with small quantities of carbon dioxide and nitrogen and traces of oxygen and other gases.

Illuminating gas is now usually burned, when used as an illuminant, in a special form of lamp in which the flame is used to heat to incandescence a fabric consisting of oxides of thorium and cerium. The quantity of light obtained by the combustion of a given quantity of gas in such a lamp is many times greater than would be obtained by the combustion of the same quantity of gas in an open flame. The illuminating power of the lamp depends only on the heating value of the gas and not upon its illuminating power when burned in an open flame. Illuminating gas companies are now obliged by law in most places to manufacture gas having a given candle-power when burned in an open flame under standard conditions. It is to be hoped that this requirement will speedily be abolished, since the addition of illuminants to water gas is an expensive process and is of no practical value at the present time. Should the candle-power requirement be abolished, water gas could be made and sold at a price which would make it practicable to employ gas as an industrial fuel for a great many purposes for which coal is now used. For instance it would then be practicable to substitute small gas engines for electric motors or isolated steam plants, and gas for electricity in the illuminations of factories, to employ gas as a fuel in ovens, furnaces and forges, and even to employ it for domestic heating.

303. Natural Gas. Natural gas is a gas which is obtained from petroleum, bearing strata of rock, at considerable depths in the earth's crust. It is obtained by drilling deep wells, and usually flows from porous sandstone rock saturated with petroleum in which the natural gas is dissolved under pressure. When the pressure is relieved by the drill, the gas evaporates from the liquid in which it has been dissolved and escapes from the openings, under a pressure ranging from a few ounces to many hundreds of pounds per square inch. Natural gas consists almost

entirely of marsh-gas together with traces of hydrogen, hydrocarbons, carbon dioxide, oxygen and nitrogen. Its heating value ranges from 900 to 1000 B.T.U. per cubic foot. Its price usually ranges from 10 to 40 cents per thousand cubic feet, and it forms an ideal fuel when it is available. Natural gas is often transported long distances in pipe lines. The city of Cleveland, for instance, is supplied with natural gas from West Virginia. The fall in pressure in this long pipe line is so great that the gas must be compressed to a pressure of several hundred pounds per square inch before delivering it to the line. This is done by means of large gas compressors which are driven by gas engines. The supply of natural gas, however, seems to be limited, so that it is gradually failing in many fields and it is not, therefore, a fuel of as great commercial importance as might be expected.

304. Producer Gas. Producer gas is a fuel which is made by the partial combustion of coal or coke with air containing an excess of moisture. The simplest form of apparatus for this purpose is that illustrated in Fig. 171. *A* is a refractory cylinder containing a quantity of broken coke heated to incandescence. The cylinder may be of steel plating lined with firebrick, or it may be a water-jacketed cast-iron cylinder. At the bottom of the cylinder there is usually a grate upon which the fuel rests. At the top is a hopper which is closed by means of a cone in the manner shown. The cylinder is charged by filling the hopper with coke and then depressing the cone, which allows the fuel to drop into the producer, after which the cone is raised and bears against the hopper, forming a gas-tight joint. The depth of fuel is usually from 3 to 5 feet or more. Some form of blower or suction apparatus forces or draws air through the fire. As the air passes through the lower layers of fuel, its oxygen unites with carbon to form carbon dioxide. When the oxygen is practically eliminated, carbon monoxide begins to be formed by the action of the incandescent carbon upon the carbon dioxide, the reaction being

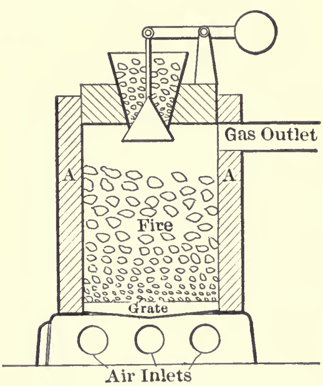


FIG. 171.—Diagram of a gas producer.

$$CO_2 + C = 2CO.$$

The gas which finally comes from the fire therefore consists almost entirely of nitrogen and carbon monoxide, when the air used in blowing the producer is free from water vapor.

Air consists of 79.3 per cent of nitrogen and 20.7 per cent of oxygen by volume. Each volume of oxygen becomes in the producer two volumes of carbon monoxide, so that one volume of air becomes 1.207 volumes of producer gas. Of this producer gas, $\frac{79.3}{120.7}$ or 65.7 per cent by volume is nitrogen, which has no heating value. The remainder, or 34.3 per cent, however, is carbon monoxide, which has a heating value of 338 B.T.U. per cubic foot under standard conditions (i.e., under a pressure of one atmosphere, and at a temperature 32° F). It will be seen, then, that such a producer gas will have a heating value of about 116 B.T.U. per cubic foot.

In burning a pound of carbon to CO, $1\frac{1}{3}$ pounds of oxygen or 5.83 pounds of air are required and 4400 B.T.U. are liberated. The water equivalent of the 2.33 pounds of carbon monoxide and 4.50 pounds of nitrogen formed as a result of the combustion is 1.67. The rise in temperature as a result of the combustion will therefore be

$$\frac{4400}{1.67} = 2640° \text{ F.},$$

which will be the approximate temperature of the gas coming from the producer. This temperature is much higher than is usual or desirable, and the sensible heat of the gas coming from the producer is lost unless this gas is used immediately for heating purposes, without cleaning or cooling it. The heat of combustion of 1 pound of carbon is 14,500 B.T.U. The heat of combustion of the $2\frac{1}{3}$ pounds of carbon monoxide formed from this pound of carbon is $2.33 \times 4380 = 10{,}200$ B.T.U. It will be seen that if the gas is cooled, the efficiency of the producer will be

$$\frac{10{,}200}{14{,}500} = 70.3 \text{ per cent.}$$

In order to increase the efficiency of the producer and also in order to increase the heating value of the gas formed, it is desirable to introduce a quantity of water vapor with the air which enters the producer. The water vapor attacks the carbon in the fire, forming carbon monoxide and hydrogen according to the reaction.

$$\begin{array}{ccccccc} C & + & H_2O & = & CO & + & H_2 \\ 12 & & 18 & & 28 & & 2 \end{array}$$

The effect of this chemical reaction is to cool the fire, since the reaction absorbs heat. The quantity of water vapor admitted with the air must be so adjusted that the temperature of the gases coming from the fire shall be that which will cause the producer to operate in the most efficient

and satisfactory manner. This requires a temperature of about 1800° F., which means that the producer gas carries with it, as it leaves the producer, a considerable amount of sensible heat. This heat will be lost unless it is utilized to evaporate the water vapor used in the producer, and to preheat the air supplied to it. If the temperature of the gas coming from the producer could be reduced to the temperature of the external air, the only loss which would be sustained as a result of the operation of the producer would be the radiation loss, which, by proper design, may be made very small. On account of the radiation loss, and the inefficiency of the regenerative apparatus which must be employed in preheating the air and evaporating the water vapor used, the efficiency of the producer cannot usually be greater than from 85 to 90 per cent, and in most practical cases it is below rather than above 85 per cent. Assuming a producer efficiency of 85 per cent, we will find that the heating value of gas produced per pound of carbon burned will be

$$14{,}500 \times 0.85 = 12{,}300 \text{ B. T.U.}$$

Assume that of each pound of carbon burned, x pounds unite with the oxygen from the air to form air-producer gas, and $1-X$ pounds react with steam to form water gas. In forming water gas, 1 pound of carbon forms $1/6$ of a pound of hydrogen, whose heating value is 10,300 B.T.U., and 2.33 pounds of carbon monoxide, whose heating value is 10,200 B.T.U. The heating value of the water gas produced by 1 pound of carbon is therefore

$$10{,}300 + 10{,}200 = 20{,}500 \text{ B.T.U.}$$

In forming air-producer gas, 1 pound of carbon forms $2\frac{1}{3}$ pounds of carbon monoxide, whose heating value is 10,200 B.T.U. We have already seen that the efficiency of the producer is such that the heating value of the gas produced from 1 pound of carbon will be 12,300 B.T.U. Hence we may write the equation

$$10{,}200X + 20{,}500(1-X) = 12{,}300.$$

Solving this for X we will find that of each pound carbon burned in the producer, 0.796 pounds unite with the oxygen of the air to form CO, and 0.204 pounds react with water vapor to form CO and H. The gas formed from 1 pound of coke will therefore, consist of $2\frac{1}{3}$ pounds of CO, $4.50 \times 0.796 = 3.59$ pounds of nitrogen and $0.204 \times \frac{1}{6} = 0.035$ pounds of hydrogen. The volume of this gas under standard conditions will be

$$\frac{2.33}{0.0781} + \frac{3.59}{0.0783} + \frac{0.035}{0.00559} = 82.0 \text{ cu.ft.}$$

The heating value of the gas per cubic feet will be

$$\frac{12,300}{82} = 150 \text{ B.T.U.}$$

It will be seen that the more efficient the producer the larger will be the amount of hydrogen and carbon monoxide contained in the gas, the less the amount of nitrogen, and the higher the heating value of the gas per cubic foot. In practice, producer gas always contains a small per cent of CO_2 and a still smaller per cent of free oxygen.

305. Auxiliary Apparatus Employed with the Gas Producer. The air from a producer may be drawn or forced through the fire in several ways. In case the air is supplied by means of a steam blower or a fan, the producer is termed a pressure producer. In case it is drawn through

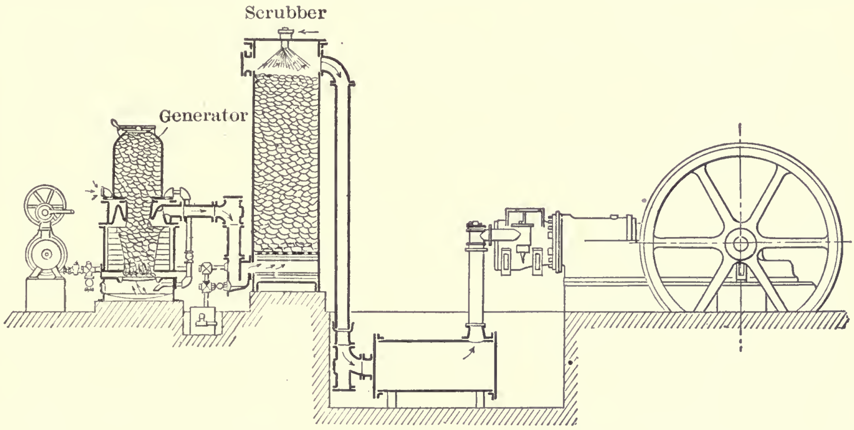


Fig. 172.—Suction gas producer plant.

the producer by the suction of a gas engine or by means of a fan, it is called a suction producer. In the first case the pressure of the gas in the producer is greater than that of the atmosphere. On account of the great depth of the fire in the producer, a considerable difference of pressure is usually required in order to operate it, the difference ranging from 3 to 8 inches of water. After the gas comes from the producer, it must be cleaned and cooled if it is to be stored, or to be used in a gas engine, or to be transmitted by means of iron piping. If, however, the producer gas is to be used immediately for heating, as, for instance, in an open-hearth furnace in a steel works, it is not necessary to clean and cool it. The cleaning of producer gas is usually accomplished by passing it through an apparatus termed a scrubber, which is a vertical cylinder filled with broken stone, coke, wooden checker work or something of that kind through which a stream of water is caused to trickle. As the current of

gas ascends through the scrubber, the action of the wet surface is, of course, to cool the gas, to extract the dust and condensible vapors from it, and finally to allow it to escape at the temperature of the entering scrubbing water, and in a form suitable for use in a gas engine or other apparatus. A suction gas producer equipped with a scrubber and attached to a gas engine is shown in Fig. 172.

306. Coal in Gas Producers. Gas producers are usually supplied with coal as the fuel from which the gas is made. The combustible in anthracite coal is nearly pure carbon, and the quality of gas made when it is used is practically the same as that which would be made from coke. However, one difficulty is encountered in the operation of an anthracite gas producer which would not be encountered in the case of a producer operated with pure carbon. Anthracite coal contains a considerable percentage of ash, and in some cases the ash is of such a nature that it fuses at the temperature encountered in the producer, and forms a vitrified mass termed clinker. The presence of quantities of clinker has a very bad effect upon the operation of the producer. If a coal is to be used which contains a considerable quantity of ash likely to clinker, the producer must be especially designed to permit of the ready removal of the clinker and so far as possible, it must be operated in such a way as to prevent its formation. A high producer temperature favors the formation of clinker. The free use of steam prevents the formation of clinker by lowering the temperature of the fire, especially of those parts of the fire which contain the largest proportion of ash. It is therefore necessary to be liberal in the use of steam if the clinker is not to become exceedingly troublesome.

In the case of bituminous coal, two other difficulties are encountered in producer operation. The first is due to the fact that the coal fuses in the fire, forming huge masses of coke through which the air cannot force its way. It is therefore necessary to break up the coke by poking the fire at intervals in the case of a bituminous producer. Bituminous coal sometimes clinkers, although usually less trouble is encountered from clinkering in operating a bituminous producer than in operating an anthracite producer.

The principal difficulty encountered in operating a bituminous producer comes from the fact that the volatile matter which is disengaged from the coal in the producer consists very largely of tar and condensible gases. If the gas is to be used immediately for heating purposes, no trouble results on this account. If, however, the gas is to be employed as a gas-engine fuel, or if it is to be stored and distributed in iron pipes, this is a serious matter. Two methods are available for overcoming this difficulty. The first one consists in condensing the tar and separating it from the gas, as is done in the case of illuminating gas. This method is not a

satisfactory solution of the difficulty, however, as the separation of the tar from the gas involves a loss in heating value and producer efficiency, and is also an expensive process, since the tar recovered usually has little or no commercial value. A preferable method is to cause the gases from the producer to pass through the fire so that the tarry vapors are decomposed into permanent gases by the action of the heat and steam. This method is less expensive than the mechanical extraction of the tar, and the producer efficiency is higher on account of the presence in the gas of the substances resulting from the decomposition of the tar.

Two types of producers are employed for fixing the tar. In the first type the gas is taken from the producer at or near the hottest point. Such a producer is shown in Fig. 173. The coal is introduced at the top of the apparatus and moves downward as it burns, the ash being taken from the bottom. Air and steam are introduced both at the top and at the bottom of the producer. The gas is taken from an opening near the center of the producer. As it moves downward through the upper half of the producer, the fuel is coked and its volatile matter distilled. The products of distillation, together with steam and air, pass downward through the bed of incandescent coke at the center of the producer, where the action of the heat and of the steam decomposes the tarry vapors, transforming them into permanent gases.

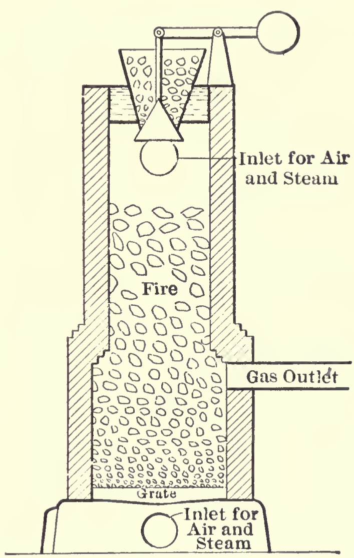


Fig. 173.—Section of a bituminous coal producer for the elimination of tar.

In the second type of bituminous producer the apparatus is divided into two or more retorts. The gases coming from the first retort pass through the second retort before they are taken to the cleaning apparatus. In such a producer one of the retorts always contains a fresh fire. The gases from this retort contain an excess of air and steam. On passing into the next retort they are drawn through a bed of incandescent coke. The effect of the heat and the excess of air and steam is to fix the tarry vapors, which have been distilled from the coal in the first retort. The gases may be then drawn through a third or fourth retort in a similar manner. One retort out of the group will usually be out of commission for the purpose of cleaning it and rebuilding a fresh fire.

307. Blast-furnace Gas. The blast furnace is an approximately cylindrical retort in which iron ore is reduced by the action of carbon. The furnace is operated by filling it to a considerable depth with hot coke, and upon this bed of coke placing successive layers of iron oxide, limestone and coke. In passing through the deep bed of coke, the air which is used to blow the furnace is transformed into nitrogen and carbon monoxide. The carbon monoxide subsequently reacts with the iron oxide and is partially transformed into carbon dioxide. The gas formed is similar to air-producer gas, except that the content of carbon dioxide is quite high. Blast-furnace gas usually consists of about 26 per cent of carbon monoxide, 3 per cent of hydrogen, 10 per cent of carbon dioxide, 56 per cent of nitrogen and about 5 per cent of water vapor and hydrocarbons. Its heating value is about 98 B.T.U., per cubic foot. About 140,000 cubic feet of blast furnace gas are produced for each ton of iron produced in the furnace. This amount of gas is more than sufficient to preheat the air admitted to the furnace, and to furnish all the power required by the furnace and the steel plant usually associated with it. It will be seen that every blast furnace is thus a source of power.

The principal difficulty encountered in the use of blast-furnace gas in gas engines arise from the fact that the gas is laden with dust from the ore. This dust is destructive to the rubbing surfaces of the cylinder, and collects upon the clearance surface, causing pre-ignition of the charge. Before the gas is suitable for use in a gas engine, the dust must be removed by the use of fans and scrubbers. Several types of patented devices are at present on the market for cleaning blast-furnace gas, and most of them are very efficient. A properly cleaned blast-furnace gas is an ideal internal combustion fuel, and engines using it have given very high efficiency.

308. Other Fuel Gases. Other fuel gases are used sometimes for special purposes in the arts. For instance, acetylene is much used for lighting, and in connection with oxygen for welding and metal-cutting operations. The cost of a given quantity of heat when obtained by the combustion of such gases is, however, so much higher than when obtained by the combustion of ordinary fuel gases, that they are never employed except when superior convenience or some similar consideration dictates their use in special cases. They are usually generated by special processes from comparatively high priced chemicals, and the methods employed are those which are especially adapted for the generation of small quantities of gas.

PROBLEMS

1. Illuminating gas having a heating value of 600 B.T.U. per cubic feet, is sold at \$0.80 per thousand cubic feet. What is the cost per million B.T.U.?

Ans. \$1.33⅓

2. What is the cost per million B.T.U. of heat from coal having a heating value of 13,000 B.T.U. per pound, and selling at \$3.50 per ton of 2000 lbs.? Ans. \$0.13½

3. Natural gas having a heating value of 1000 B.T.U. per cubic foot is sold at \$0.25 per thousand cubic feet. What is the cost per million B.T.U.? Ans. \$0.25

4. Producer gas having a heating value of 125 B.T.U. per cubic foot is sold at 6 cents per thousand cubic feet. What is the cost per million B.T.U.? Ans. \$0.48

5. A gas producer employing pure carbon as a fuel has an efficiency of 100 per cent. What will be the heating value of the gas produced per pound of carbon?

Ans. 14,500 B.T.U.

6. What proportion of the carbon reacts with the air to form carbon monoxide?

Ans. 58.2%.

7. What proportion reacts with the steam to form carbon monoxide and hydrogen?

Ans. 41.8%.

8. How many pounds of carbon monoxide will be formed per pound of carbon burned? Ans. 2.33 lbs.

9. How many pounds of hydrogen will be formed per pound of carbon burned?

Ans. .0697 lbs.

10. How many pounds of nitrogen will be in the gas produced per pound of carbon? Ans. 2.59 lbs.

11. What will be the volume of the gas produced per pound of carbon?

Ans. 75.2 cu.ft.

12. What will be its heating value per cubic foot? Ans. 193 B.T.U.

CHAPTER XXII

COMPRESSED AIR

309. The Air Compressor. Air under high pressure, as an agent for the transmission of power, has a very wide application in certain industries, notably in quarrying, mining, and in railway work. Air under high pressure is usually termed **compressed air,** and is obtained by the use of an apparatus termed an air compressor. The type of air compressor most usually employed is that known as a piston compressor, and consists of a cylinder provided with suitable valves and within which there is a reciprocating piston. The air compressor is in effect a pump which differs from an ordinary water pump only in the fact that it handles an elastic instead of an inelastic fluid. It is therefore feasible to operate the air compressor at much higher speeds than a water pump, and it is necessary to make certain changes in the design of the mechanism on account of the nature of the fluid pumped.

A section through an air compressor cylinder is shown in Fig. 174. As the piston moves to the left, air is drawn into the cylinder at atmospheric pressure and temperature, through the inlet valve I. When the piston reaches the end of its stroke, the spring shown automatically closes the valve. During the return stroke the air undergoes polytropic compression, but since the amount of heat transferred to the wall of the cylinder (which is water-jacketed) is small, the compression is practically adiabatic. When the pressure of the air in the cylinder becomes equal to that in the receiver into which the air is to be discharged, the discharge valve *D* opens and permits the air to flow into the receiver. It is, of course, impossible to make an air pump without any clearance volume. It will therefore be seen that at the end of the discharge stroke, the cylinder will contain a small quantity of air at high pressure. As the piston moves forward, the pressure of this air will hold the inlet valve closed, and it

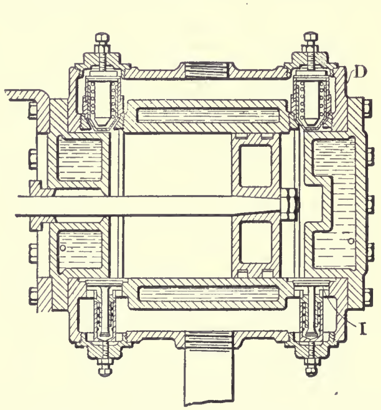


Fig. 174.—Section of an air-compressor cylinder.

will expand adiabatically to a considerable volume before its pressure falls to that of the atmosphere. As soon as the pressure falls to atmospheric pressure, a fresh supply of air will be drawn through the inlet valve.

310. Form of the Air-compressor Card. The form of the card theoretically given by an air compressor of the piston type is shown in Fig. 175. The cylinder is filled with air at atmospheric pressure and temperature at point *a*. The line *a-b* represents the adiabatic compression of the air. The pressure represented by the ordinate to the point *b* is that of the air in the receiver into which the air is to be discharged. During the remainder of the compression stroke the air is discharged at constant pressure as represented by the line *b–c*. When the piston makes its return stroke, the air contained in the cylinder at point *c* expands adiabatically, the expansion line being *c–d*. While the piston is moving from *d* to point *a* the supply of air for the next stroke is being drawn through the inlet valve, the line *d–a* being the induction line.

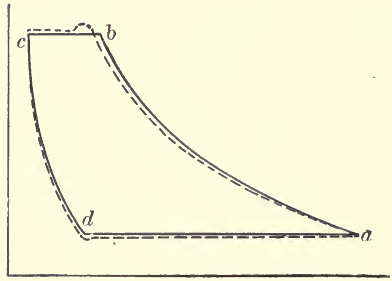


Fig. 175.—Theoretical card from an air compressor.

The form of card which would actually be given by an air compressor is shown by the dotted lines in the same figure. During the compression period, on account of the heat lost to the water-jacket, the compression line falls somewhat below the adiabatic line. Before the air can be expelled from the cylinder, however, its pressure must rise above that of the receiver, since a difference in pressure is necessary in order to lift the discharge valve against the resistance of its springs and to force the air through the constricted passage through which it must be delivered. In consequence, the actual discharge line lies above the theoretical discharge line. Since the discharge valve must be opened quickly in order to allow the air to escape, an extra difference in pressure is necessary at first on account of the inertia of the valve. The beginning of the discharge line therefore usually has the form shown. In case the weight of the valve and the

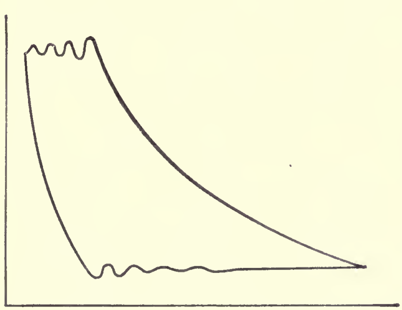


Fig. 176.—Effect of fluttering of valves on an air compressor card.

strength of the spring have a certain relation, the current of air will cause the valve to vibrate, which will produce a series of waves in the discharge line as shown in Fig. 176. During the early part of the suction stroke, while the air contained in the clearance is expanding, it is losing heat to the water-jacket, so that the expansion line falls somewhat below the theoretical line. At the beginning of the suction period, as at the beginning of the discharge period, an extra difference in pressure is needed in order to open the inlet valve. The suction line, therefore, has the form shown, and in case the relation between the weight of the valve and the strength of the spring will permit of it, the suction line will also have a series of waves as shown in Fig. 176. As a result of the friction of the air in passing through the valves and of the difference in pressure necessary to open the valves, the work required by the compression in somewhat greater than that theoretically required to adiabatically compress the air and deliver it to the receiver. In consequence of fluid friction and of the pressure required to open the inlet valves, the volume of air taken into the cylinder, and therefore the capacity of the compressor, is reduced. It is therefore advisable in designing compressors to so proportion the valves as to make the losses from these sources a minimum.

311. The Multi-stage Compressor. The air which is discharged into the receiver from the cylinder is of high temperature on account of its adiabatic compression. When this air enters the piping system, it loses heat by conduction and radiation, and shrinks in volume. Since the air is subsequently cooled, it will be apparent that the heat imparted by compression will be lost, and that it will take less work to compress and deliver a given volume of compressed air if the air is compressed isothermally instead of adiabatically. Hence, when air is wanted at high pressures, it is usual to employ what are termed two-stage compressors (i.e., compressors in which the air is compressed to some intermediate pressure in a large cylinder, is then delivered to an intermediate receiver in which it is cooled to approximately its initial temperature, and then compressed in a second small cylinder to its final pressure). When very high pressures are required, three-stage and sometimes four-stage compressors are employed. The cards which would be given by such a compressor are shown in Fig. 177. The adiabatic compression line for the first cylinder is the line a–b. The discharge line for the first cylinder is

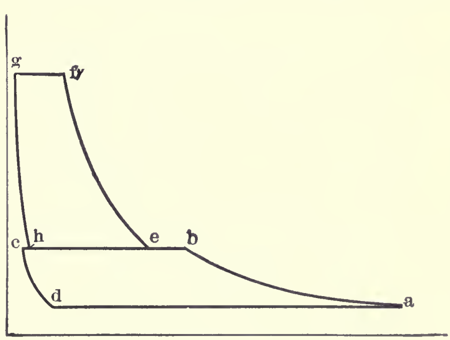


Fig. 177.—Cards from a two-stage compressor.

the line b–c. The expansion and suction lines for this cylinder are c–d and d–a. In consequence of the cooling which the air undergoes in the intermediate receiver, the volume of the air discharged per stroke of the compressor is reduced from the volume represented by the line c–b, to that represented by the line $h\ e$. In the second cylinder the air is again compressed, the compression line $e\ f$ being practically adiabatic. The discharge line in the high-pressure cylinder is the line $f\ g$, and the air contained in the clearance space expands along the line $g\ h$. In Fig. 178, a theoretical card from a two-stage compressor without clearance is superimposed upon the cards which would be given with adiabatic compression and with isothermal compression. The isothermal compression line is the line, $a\ c\ e$ the adiabatic compression line is the line $a\ b\ g$ and the theoretical cards are shown in full lines. It will be seen that the area $a\ b\ c$ + the area $c\ d\ e$ is work lost on account of the fact that the compression in the cylinders is adiabatic and not isothermal. Area $c\ b\ g\ d$ represents the work saved by the employment of a two-stage compressor.

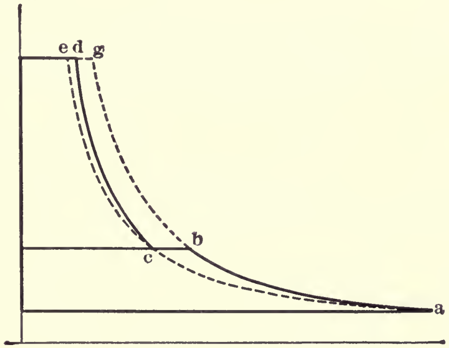


FIG. 178.—Theoretical cards from a two-stage compressor.

312. The Work of Compression. In theory, the least quantity of work necessary in order to compress a given quantity of air and to deliver it into a receiver, is that quantity of work required to compress it isothermally. Referring to Fig. 179, which is the ideal card given by an air compressor without clearance and having isothermal compression, it may be seen that the amount of work performed by the air in entering the cylinder is represented by the area $o\ d\ a\ a'$. The amount of work required to compress the air isothermally is that represented by the area $b'\ b\ a\ a'$. The amount of work required to deliver this air into the receiver is measured by the area $o\ c\ b\ b'$. Let P_a be the pressure of the atmosphere in pounds per square foot, V_a be the volume of one pound of air at atmospheric pressure and temperature and P_r the pressure of the air in the receiver.

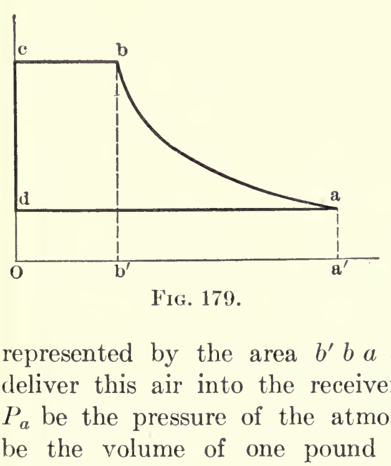


FIG. 179.

Then the work done by the air entering the cylinder will be

$$U_a = P_a V_a = R T_a, \quad \ldots \ldots \quad (1)$$

The ratio of compression will be

$$r = \frac{P_r}{P_a}. \quad \ldots \ldots \quad (2)$$

The volume of the air after compression will be

$$V_r = \frac{V_a}{r}. \quad \ldots \ldots \quad (3)$$

The work of compression will be

$$U_c = P_a V_a \log_e r = R T_a \log_e r, \quad \ldots \ldots \quad (4)$$

the work required to deliver the air into the receiver will of course be

$$U_d = P_r V_r = P_a V_a = U_a \quad \ldots \ldots \quad (5)$$

From this it will be seen that the work required to compress and deliver the air will be

$$U = U_c + U_d - U_a = P_a V_a \log_e r \quad \ldots \ldots \quad (6)$$

from which it may be seen that the work required to compress the air and deliver it is equal to the work of isothermal compression.

The efficiency of compression of a compressor may be defined as the ratio of the work theoretically required for isothermal compression, to the indicated work actually shown by the cards from the compressor, to be necessary to compress and deliver a given quantity of air. It will be seen that the more nearly isothermal the compression is, the higher will be the efficiency of the compressor. Since the multi-stage compressor approximates isothermal compression more closely than does the single-stage compressor, the efficiency of compression in the case of a multi-stage compressor is much higher than in the case of a single-stage compressor. The efficiency of a compressor is also increased by making the valve passages of ample area and so designing the valves that the excess pressure required to open them is a minimum.

The work of adiabatic compression may be seen by referring again to Fig. 178. The compression line may now be assumed to be adiabatic instead of isothermal. The work done by the air in entering the cylinders is as before

$$U_a = P_a V_a = R T_a \quad \ldots \ldots \quad (7)$$

If P_r equals the pressure in the receiver against which the air is to be discharged, we will have for the volume of the air at point b a quantity in which we can designate by the symbol V_r. For its value we will have

$$V_r = V_a\left(\frac{P_a}{P_r}\right)^{\frac{1}{\gamma}}. \qquad (8)$$

The work done in adiabatically compressing the air which is represented by the area $b'\ b\ a\ a'$, is

$$U_c = \frac{P_r V_r - P_a V_a}{\gamma - 1}. \qquad (9)$$

Substituting the value of V_r from equation (8), we have

$$U_c = \frac{V_a\left(P_a^{\frac{1}{\gamma}} P_r^{\frac{\gamma-1}{\gamma}} - P_a\right)}{\gamma - 1}. \qquad (10)$$

The work of delivering the air into the receiver is

$$U_d = V_r P_r = V_a P_a^{\frac{1}{\gamma}} P_r^{\frac{\gamma-1}{\gamma}}, \qquad (11)$$

the net work of adiabatic compression will therefore be

$$U = \frac{\gamma V_a}{\gamma - 1}\left(P_a^{\frac{1}{\gamma}} P_r^{\frac{\gamma-1}{\gamma}} - P_a\right). \qquad (12)$$

As a usual thing the actual work of compression will be somewhat greater than this.

313. Losses Due to Clearance and Altitude. The indicated power lost on account of the presence of air in the clearance space of the compressor at the end of the discharge period is inconsiderable in amount, since the air performs almost as much work during its expansion as was performed upon it during its compression. Were there no heat transfer to the cylinder walls, there would be no loss of power from clearance. However, the use of clearance necessitates the employment of a compressor cylinder whose swept volume is considdrably larger than the volune of the free air compressed per stroke. On account of this increase in size of the compressor cylinder, the friction loss in the compressor will be larger and the first cost of the machine will be greater than they would be if a cylinder without clearance were used. The use of large clearance in connection with an air compressor is therefore undesirable and should be avoided as far as possible.

The ratio of the quantity of the free air actually taken in per stroke, to the swept volume of the air compressor is termed the volumetric

efficiency of the compressor. A high volumetric efficiency is desirable for the reasons already indicated. This is particularly the case when the compressor is to be operated at high altitudes. At considerable elevations the pressure of the air is much reduced, the pressure dropping off approximately at the rate of one-half pound absolute for each thousand feet in elevation above the sea level. The effect of this reduction in the initial pressure of the air is to decrease the weight of a given volume of free air, and therefore, the decrease in volume of compressed air delivered by each stroke of the piston. Consequently larger compressors are required at high altitudes than at sea level, and the volumetric efficiency of the compressor at high altitudes is less than it is at sea level. Compressors are usually rated by the number of cubic feet of free air which they will deliver per minute at normal speed. It is, not however, the number of cubic feet of free air, but the number of cubic feet of compressed air required which determines the size of the compressor. At high altitudes, therefore, it is an important matter to make certain that the compressor is of sufficient capacity.

314. Mechanically Operated Valves. When an air compressor equipped with automatic valves is operated at high speed, it becomes necessary to provide them with rather stiff springs, in order to insure that they shall close promptly and avoid waste of air. The use of such springs, however, causes a waste of power and a reduction in the capacity of the machine, as was seen when comparing the actual air compressor diagram with the theoretical diagram in Fig. 175. The stiffer the springs with which the valves are equipped, the greater the difference in pressure which will be required in order to cause them to open. In order to avoid the loss in power and capacity resulting from the use of automatic valves, the larger size of air compressors are often equipped with semi-rotary valves similar to the valves used in a Corliss engine, the inlet valves being similar in form to Corliss exhaust valves and the discharge valve similar in form to Corliss inlet valves. In many cases inlet valves of the Corliss type are combined with automatic discharge valves.

315. Blowing Engines. An air compressor which delivers air under a pressure of from 15 to 30 pounds for use in blast furnaces and steel works, is usually termed a blowing engine, and the cylinder of such a compressor is termed a tub. The principles of operation of blowing engines are the same as those of other air compressors. Blowing engines are usually fitted with mechanically operated valves, instead of automatic valves, since they are commonly operated at high speed on account of their large capacity. However, several builders are now equipping blowing engines with automatic valves made of thin sheet steel which are held against their seats by very light springs. Since blowing engines deliver air against comparatively low pressure, the losses due to the heating of

the air by adiabatic compression are comparatively small, and those due to fluid friction and the imperfection of the valve action are comparatively large. Consequently, blowing engines are made single stage, and a great deal of care is taken in designing the valves.

316. Moisture in Compressed Air. The principal difficulties encountered in the use of compressed air arises from the moisture which is contained in the air. Air which is compressed to a pressure of 80 pounds gage, is reduced to about 15 per cent of its former volume. Consequently a given mass of it can contain at a given temperature only about 15 per cent of the moisture which it was able to contain previous to compression. Hence if the humidity of the air is greater than 15 per cent, some of the moisture will be deposited as water in the receiver and piping system. Usually, from 50 to 80 per cent of the moisture contained in the air is deposited in the piping system on this account and it gives a great deal of trouble, particularly in winter, from freezing. In order to reduce the trouble from this source, it is customary to cool the air coming from the air compressor, in order that the moisture which it contains may be deposited in the receiver and removed before the air enters the piping system. This process is known as after-cooling. In order to reduce the quantity of moisture in the air, and also in order to reduce the amount of work required to compress it, it is advisable to take the supply of air for the compressor from the coolest point possible.

317. Example of Air-compressor Design. The following example will serve to show the method of calculating the size of air-compressor cylinders. Assume that a compressor is required which will deliver 100 cubic feet of compressed air per minute at a gage pressure of 120 pounds. The normal pressure of the atmosphere will be assumed to be 13 pounds per square inch and the atmospheric temperature to be 60°. The absolute pressure of compression will be 133 pounds per square inch. The ratio of the final to the initial pressure will be $133 \div 13 = 10.2$. The number of cubic feet of free air required per minute will therefore be 1020. If the compressor is to operate at 60 revolutions per minute, the number of cubic feet of free air compressed per stroke must be 8.53. In order to equalize the work done in the two cylinders of a two-stage compressor, the ratio of compression should be the same in each one. The ratio of compression in each cylinder will therefore be

$$r = \sqrt{10.2} = 3.2.$$

The pressure of the air in the intermediate receiver will therefore be $13 \times 3.2 = 41.5$ lbs. The volume of the air will be

$$V_r = V_a \left(3.2\right)^{\frac{1}{\gamma}} = .438\, V_a.$$

Assuming that the clearance volume of the cylinder is 4 per cent of the swept volume, we will have for the volume of the air contained in the cylinder at the beginning of the suction period, $4 \div 0.438 = 9.14$ per cent of the swept volume. Since the total volume of the cylinder is 104 per cent of the swept volume, the volume of the free air taken in per stroke will be $104 - 9.14 = 94.86$ of the swept volume of the cylinder. The swept volume of the cylinder must therefore be

$$8.53 \div .949 = 9 \text{ cubic feet.}$$

Assuming the stroke of the compressor to be 4 feet, the diameter of the low-pressure cylinder will be $20\frac{1}{4}$ inches. In practice, this diameter would be somewhat increased.

With the same ratio of compression and the same clearance volume in the high-pressure cylinder the swept volume of the high-pressure cylinder will be equal to the swept volume of the low-pressure cylinder divided by the ratio of compression. Consequently, the diameter of the high-pressure cylinder will be

$$20\tfrac{1}{4} \div \sqrt{3.2} = 11\tfrac{3}{8} \text{ inches.}$$

The valves of the compressor are usually designed so that the nominal velocity of the air passing through them will be 6000 feet per minute. In order to prevent the cumulative action which would result from the heating of the cylinder walls by the adiabatic compression of the air, a water-jacket must be provided. Were it not for this water-jacket, the cylinder walls would become heated by the air and they in turn would heat the entering air. The adiabatic compression of this heated air would still further heat the walls and the result would be that both the temperature of compression and the temperature of the cylinder walls would increase together until radiation from the walls would balance the heat received from the air. As a result, a large quantity of power would be required for the operation of the compressor, its volumetric efficiency would be seriously reduced, and it would be impossible to lubricate the rubbing parts.

318. Flow of Air or Gas in a Tube. When a fluid is caused to pass through a tube it is found that a difference in pressure is required at the two ends of the tube in order to cause the passage of the fluid. The amount of this difference in pressure is usually much greater than that which is required in order to give the fluid the actual velocity which it has in the tube. We are therefore obliged to conclude that there is a force analogous to friction opposing the flow of the fluid, and that the work done by this force is transformed into heat and raises the temperature of the fluid. Experiment shows this to be the case. It further shows that the amount of this force is proportional to the length of the tube, and to the 1.8 power of the velocity of the fluid and inversely proportional to the 1.3 power of the diameter of the tube. It shows that the force

depends upon the character of the walls of the tube, and is greater in the case of tubes having rough walls and less in the case of smooth walls. It shows that the force is proportional to the density of the fluid and also to a property which we term the viscosity of the fluid. Glycerine, for instance, is not greatly denser than water, and yet we know by experience that it is thicker or more viscous and we find that the force of fluid friction is much greater in the case of glycerine than in the case of water. We may express these observed facts by the equation

$$\frac{dP}{dL} = K\,S\,\frac{v^{1\cdot 8}}{d^{1\cdot 3}}, \quad \ldots \ldots \ldots \quad (1)$$

in which dP is the difference in pressure in pounds per square foot, between two points in a tube through which a fluid is flowing, dL is the distance of these points from one another, in feet, S is the density of the fluid in pounds per cubic foot, v is the velocity of the fluid in feet per second, d is the diameter of the tube in feet, and K is a constant depending on the character of the interior surface of the tube and also upon the viscosity of the fluid.

In the case of a gas or vapor, the density depends upon the temperature and pressure of the fluid, and since experiment shows that the viscosity of all gases is practically the same we may write for the above equation

$$\frac{dP}{dL} = K\left(\frac{P}{R\,T}\right)\frac{v^{1\cdot 8}}{d^{1\cdot 3}}, \quad \ldots \ldots \ldots \quad (2)$$

in which P is the pressure of the gas in pounds per square foot, T is its absolute temperature, R is the function $\frac{P\,V}{W\,T}$, and K is a factor which depends upon the character of the internal surface of the tube.

If we let W = the number of pounds of gas passing a given cross-section of the tube per second, then the volume of this gas will be given by the expression

$$V = \frac{W\,R\,T}{P} \quad \ldots \ldots \ldots \quad (3)$$

The area of the cross-section of the tube is equal to $\frac{\pi d^2}{4}$. The velocity of gas in the tube is found by dividing the volume of gas passing in a given time by the area of the tube, hence we may write

$$v = \frac{4}{\pi}\left(\frac{W\,R\,T}{P\,d^2}\right). \quad \ldots \ldots \ldots \quad (4)$$

Raising to the 1.8 power we will have

$$v^{1\cdot 8} = k''\left(\frac{W^{1\cdot 8}\,R^{1\cdot 8}\,T^{1\cdot 8}}{P^{1\cdot 8}\,d^{3\cdot 6}}\right) \quad \ldots \ldots \ldots \quad (5)$$

Substituting this in equation (2) we have

$$\frac{dP}{dL} = K\left(\frac{W^{1\cdot 8}\,R^{\cdot 8}\,T^{\cdot 8}}{P^{\cdot 8}\,d^{4\cdot 9}}\right). \quad \ldots \ldots \ldots \quad (6)$$

Collecting like terms we will have

$$P^{\cdot 8}\,dP = K\,W^{1\cdot 8}\frac{R^{\cdot 8}\,T^{\cdot 8}}{d^{\cdot 9}}\,dL \quad \ldots \ldots \ldots \quad (7)$$

Integrating this expression between the limits of P_1 and P, and zero and L, we will have

$$.55(P_1^{1\cdot 8} - P^{1\cdot 8}) = \frac{K\,W^{1\cdot 8}\,(R\,T)^{\cdot 8}}{d^{\cdot 9}}\,L. \quad \ldots \ldots \ldots \quad (8)$$

Which becomes

$$P_1^{1\cdot 8}-P^{1\cdot 8}=K\,W^{1\cdot 8}\frac{(R\,T)^{\cdot 8}}{d^{4\cdot 9}}\,L, \quad \ldots \ldots \ldots \quad (9)$$

in which P_1 is the initial pressure in pounds per square inch absolute at any point in the tube, P is the pressure in pounds per square inch absolute at a point L feet distant from the first point in the direction of flow, W is the number of pounds of gas passing each cross-section of the tube per second, d is the diameter of the tube in inches, R is the density function of the gas, T is the absolute temperature of the gas and K is a constant depending upon the character of the inner surface of the tube. Solving the above equation for the weight of gas transmitted per minute we will have

$$W=\sqrt[1.8]{\frac{(P_1^{1\cdot 8}-P^{1\cdot 8})\,d^{4\cdot 9}}{K\,L\,(R\,T)^{\cdot 8}}}. \quad \ldots \ldots \ldots \quad (10)$$

Solving for the diameter of the tube required to transmit a given weight of gas per minute with a given loss in pressure will have

$$d=\sqrt[4.9]{\frac{K\,W^{1\cdot 8}\,(R\,T)^{\cdot 8}\,L}{P_1^{1\cdot 8}-P^{1\cdot 8}}}. \quad \ldots \ldots \ldots \quad (11)$$

Solving for the value of the constant when it is to be determined, by experiment, we will have

$$K=\frac{(P_1^{1\cdot 8}-P^{1\cdot 8})\,d^{4\cdot 9}}{W^{1\cdot 8}\,(R\,T)^{\cdot 8}\,L}. \quad \ldots \ldots \ldots \quad (12)$$

The value of K for iron pipes is usually about 0.026.

319. Applications of Compressed Air. Compressed air may be used as the working fluid in an engine, in exactly the same way as steam is used. The card from a compressed-air motor is similar to one from a steam engine. The amount of power given by the air motor and also the weight of air used may be computed from the card. Since air is a permanent gas, there is no cylinder condensation, and the thermal loss with a compressed-air motor will be much less than with a steam engine. The power developed from a given weight of air may be increased by heating the air before it enters the motor, and if the motor is to be used continuously, it is advisable to preheat the air in this manner. It is usually advisable to use air motors in place of steam engines when compressed air is available and the motors operated for only a small portion of the time. A small steam engine is continually wasting heat when it is not in operation, and much steam is wasted by cylinder condensation in warming it up after each period of idleness. There are no such losses in the case of an air motor.

Compressed air finds its principal application in quarrying and mining, in the operation of rock drills, and channeling machines. In coal mining especially, it is impracticable to use steam for operating such machines, since the boilers must be placed above ground. To transmit steam from a boiler plant at the mouth of the mines, to engines and drills situated

underground and hundreds or thousands of feet away, would result in a very great waste of heat and in considerable danger to the workmen. When compressed air is used as the working fluid in such machines, there is no radiation of heat from the piping or losses resulting from the intermittent use of the machinery. Consequently, if the air compressor plant is efficient, the cost of operating compressed-air machinery under these conditions is much less than the cost of performing the same work by steam. Another field in which compressed air is used to great advantage is in the driving of percussion tools in shops. The pneumatic riveter which is usually employed in assembling structural work in the field is an example of such a tool. It consists of a heavy cylinder within which a small, but rather heavy piston, termed a hammer, is caused to reciprocate by the action of compressed air. A throttle valve is provided which regulates the pressure of the air admitted to the tool, and so controls the force of the blow. The hammer vibrates at a rate of several hundred strokes per minute and when provided with an extension having a face of suitable form, it rapidly batters the hot metal of the rivet into shape. Since the hammer and its extension are much lighter than the cylinder in which they are contained, the vibration of the cylinder is not so excessive but what it may be held by hand when in use. Similar, but lighter tools are employed in foundries and machine shops, where they are known as pneumatic hammers. The extension pieces which are attached to the hammers are in the form of chisels, calking tools, etc.

In work of this kind, while it is desirable that the motors which use the air shall be as economical as possible, it is very much more important that they shall be convenient to operate, shall perform their work rapidly and effectively, and shall be of such rugged construction as not to be injured by hard usage and abuse. Since these conditions are often incompatible with economy, it will be found that rock drills, pneumatic hammers and similar machinery are often inefficient, if we define the efficiency of such a piece of apparatus as the ratio of the work which it performs to the power theoretically required to compress the air which it consumes.

PROBLEMS

1. Find the quantity of work theoretically required in order to isothermally compress 10 cubic feet of free air having a pressure of 14 pounds per square inch and deliver it into a receiver in which the pressure is 70 pounds per square inch.

Ans. 32,450 ft.-lbs.

2. Find the work theoretically required to adiabatically compress 10 cubic feet of free air having a pressure of 14 pounds per square inch and deliver it into a receiver in which the pressure is 70 pounds per square inch? Ans. 41,700 ft.-lbs.

3. Find the efficiency of compression when the compression is adiabatic?

Ans. 78%.

4. A multistage compressor compresses air having a temperature of 60° F. from a pressure of 14 pounds per square inch absolute to a pressure of 56 pounds per square inch absolute in the first stage. The air is then cooled to 60° F. In the second stage, the pressure is raised from 56 pounds to 224 pounds absolute. Find the work required per cubic foot of free air compressed, assuming the compression to be adiabatic in each stage. Ans. 7110 ft.-lbs.

5. Find the work required assuming that the compression was adiabatic and was completed in one stage. Ans. 8650 ft.-lbs.

6. Find the per cent of work saved by the employment of two-stage compression. Ans. 17.8%.

7. A compressor compresses air adiabatically from a pressure of 14 pounds absolute to a pressure of 84 pounds absolute. The clearance volume is 5 per cent of the swept volume. Find the volume of the air contained in the cylinder at the beginning of the suction period, expressed as a per cent of the swept volume of the cylinder? Ans. 17.85%.

8. Find the volumetric efficiency of the compressor. Ans. 87.15%.

9. Assume that the initial pressure in Problem 7 is 10 pounds absolute. Find the volume of the air contained in the cylinder at the beginning of the suction period. Ans. 22.6%.

10. Find the volumetric efficiency of the compressor under these conditions. Ans. 82.4%.

11. What must be the swept volume of a cylinder having the volumetric efficiency obtained in Problem 10, if it is to deliver 6 cubic feet of free air per stroke. Ans. 7.3 cu. ft.

12. Air having a temperature of 80° and humidity of 70% is compressed from a pressure of 14.5 pounds per square inch absolute to a pressure of 72.5 pounds per square inch gage. What quantity of moisture will it contain per cubic foot after compression and cooling to the initial temperature? Ans. .00157 lbs.

13. How many cubic feet of free air will be required per cubic foot of compressed air? Ans. 6 cu. ft.

14. How many pounds of moisture did this quantity of free air contain? Ans. .00670 lbs.

15. How many pounds of moisture are precipitated by the compression and cooling of this quantity of air? Ans. .00513 lbs.

16. What quantity of moisture will be precipitated per day in the pipe lines of an air-compressor system compressing 100,000 cubic feet of free air per day, if the conditions are those given in Problem 12? Ans. 513 lbs.

17. Assuming that the clearance volume of the high-pressure cylinder of the air compressor in Art. 317 is 6 per cent, what will be the volume of the air contained in the cylinder at the beginning of the suction period, in terms of the swept volume? Ans. 13.7%.

18. How many cubic feet of air of a pressure of 41.5 pounds must this cylinder handle per stroke? Ans. 2.665 cu. ft.

19. What must be the swept volume of the cylinder? Ans. 2.89 cu. ft.

20. What will be the diameter of the cylinder? Ans. $11\frac{1}{2}$ ins.

CHAPTER XXIII

REFRIGERATION

320. Refrigerating Machines. A refrigerating plant is an apparatus for maintaining a low temperature in a desired region by removing heat from that region and transferring it to a region of high temperature. A refrigerating machine is the converse of a heat engine, since it transforms work into heat, and then rejects the heat into a region of high temperature. Like the heat engine, the refrigerating machine employs a working fluid and causes this working fluid to undergo a thermodynamic cycle. Since the object of refrigeration is the transfer of heat, and not the performance of work, it is customary to take as the efficiency of a refrigerating system, the ratio of the heat transferred to the work done. Since the mechanical equivalent of the heat transferred is almost always several times as great as the work done, the efficiency of a refrigeration plant is usually greater than unity. A refrigerating machine may employ as a working fluid either a gas or a vapor. On shipboard, air is usually employed as a working fluid, since the leakage of air within the confined space of a ship's engine room is not harmful. In stationary plants the vapor of ammonia is usually employed as the work ng fluid. Other vapors and gases are also employed to a considerable extent.

Refrigerating plants may be divided into four classes. Machines of the first class use a permanent gas as their working fluid. After being compressed and cooled, the working fluid is expanded adiabatically and its temperature reduced to a low value. Machines of the second class, which are called vapor-compression machines, liquefy a vapor by the application of pressure, and by the subsequent re-evaporation of this liquid under low pressures the desired temperature is obta'ned. Refrigeration plants of the third class are termed absorption plants. In such plants a volatile vapor, like ammonia, is absorbed by water or some other liquid and then driven off under high pressure by heat in such a manner that it may be subsequently cooled and condensed. It is then evaporated under low pressure, thus producing a low temperature. In apparatus of the fourth class, a gas is compressed to a very high pressure and cooled. When it is subsequently expanded the work done in separating its particles against their mutual attractions lowers its temperature

(a phenomenon already referred to in Chapter III as the Joule-Thompson effect).

321. The Air-refrigerating Machine. A machine of the first class, using air as a working fluid, is represented in Fig. 180. In cylinder A air is compressed adiabatically and then forced into the condenser coil B. By its adiabatic compression, its temperature is raised so that it is somewhat higher than the temperature of the water supplied to the condenser. In the condenser, the temperature of the air is reduced a few degrees and the air then enters the smaller cylinder C, where it expands adiabatically to its original pressure. As a result, its temperature is very much reduced. It is then discharged into the coil D, usually termed a vaporizer, where it absorbs heat at low temperature from the substance which is to be cooled. In this coil the temperature of the air is raised a few degrees, and it then enters cylinder A, where it is again compressed. The condenser B may consist of a shell filled with tubes through which cooling water circulates, similar in its general arrangement to a surface condenser. This is the form of condenser usually employed on shipboard. In stationary plants, the condenser usually consists of a coil of pipe over which water is allowed to drip. By its evaporation, this water cools the air or other working fluid contained in the pipes. The vaporizer may consist of a coil of pipe enclosed in the space to be cooled, and it usually has this form when air is used as the working fluid. When ammonia is used, however, the vaporizer pipes are usually immersed in brine, and the cold brine is then circulated through pipes in the space to be cooled.

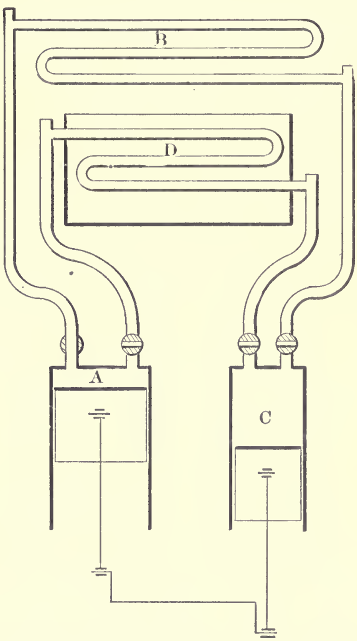


FIG. 180.—Air-refrigerating machine.

It will be seen that the machine described is an apparatus for cooling air by expansion, so that it may absorb heat from a cold body, and then heating it by compression, so that it may reject that heat to a hot body. In practice, the temperature of the air rejected by cylinder A must be considerably greater than the temperature of the water which cools it, and the temperature of the air exhausted by cylinder C must be quite a little less than the temperature of the substance to be cooled. The cycle

upon which this machine operates is the reverse of the Joule cycle which was described in Chapter XVIII.

322. Example of the Performance of an Air Machine. Assume that a machine of this type is required to maintain a temperature of 0° F., and to reject heat at a temperature of 80° F. In order to insure proper operation, we will assume that a difference of 20° is necessary to effect the heat transfer in each case, so that the air must be expanded until its temperature is −20° F. or 440° absolute, and must be compressed until its temperature is 100° F., or 560° absolute. We will assume that the cycle is performed with 1 pound of air and that the temperature of the air is raised 10° in the vaporizer. The temperature of the air entering the cylinder A will then be 450° absolute. Assume that its pressure is 14.7 pounds per square inch. The final pressure of compression may be found by the formula:

$$P_2=P_1\left(\frac{T_2}{T_1}\right)^{\frac{\gamma}{\gamma-1}}.$$

Solving we will have for the final pressure

$$P_2=14.7\left(\frac{560}{450}\right)^{\frac{1.4}{0.4}}=31.6 \text{ lbs. per sq.in.},$$

which will be the pressure of the air in the condenser. The work done during this compression may be obtained by the formula

$$R\frac{(T_2-T_1)}{(\gamma-1)}=53.2\left(\frac{560-450}{0.4}\right)=14{,}620 \text{ ft.-lbs.}$$

The work done in expelling the air into the condenser may be found by the formula

$$RT=53.2\times 560=29{,}900 \text{ foot-pounds.}$$

The temperature of the air leaving the condenser will be

$$\frac{560\times 440}{550}=547.6°$$

The work which the air does in entering the cylinder B from the condenser will be equal to

$$53.2\times 547.6=29{,}150 \text{ ft.-lbs.}$$

The work of expansion in this cylinder will be

$$53.2\left(\frac{547.6-440}{0.4}\right)=14{,}390 \text{ ft.-lbs.}$$

The work of expelling the air from the expansion cylinder will be

$$53.2 \times 440 = 23400 \text{ ft.-lbs.}$$

The work done by the air in entering the compression cylinder is

$$53.2 \times 450 = 23900 \text{ ft.-lbs.}$$

The net indicated work will be found to be 447 foot-pounds, which is the difference between the work done upon the air in cylinder A, and the work done by the air in cylinder B. The heat transferred will be found by multiplying the rise in temperature by the specific heat of air at constant pressure and will be 2.38 B.T.U. Reducing this to foot-pounds, we will have 1850 for the mechanical equivalent of the heat transferred. Dividing the heat transferred by the net indicated work, we will have 410 per cent for the efficiency of the machine. The efficiency of the Carnot refrigerating machine transferring heat from a region of 0° F. to a region of 80° F. will, of course, be

$$\frac{T_2}{T_1 - T_2} = \frac{460}{540 - 460} = 575 \text{ per cent.}$$

It will be seen that the efficiency of the reversed Joule cycle is much less than the efficiency of the Carnot cycle, although the theoretical efficiency obtained by the above computations is considerably higher than would be realized in practice. In practice, it would be found that neither the expansion nor the compression of the working fluid would be adiabatic, and in order to obtain the temperature range desired a larger pressure range would be necessary, which would increase the amount of work required to operate the machine.

It will be noted that less than $2\frac{1}{2}$ B.T.U. per cycle per pound of working fluid were transferred from the vaporizer to the condenser. In order to transfer any considerable quantity of heat by means of a refrigerating machine operating on the reversed Joule cycle, it is necessary that the machine be very large and heavy, and on account of the small amount of net work as compared with the large quantity of work performed in the two cylinders, the mechanical efficiency of the machine will be very low. In order to reduce the size of the cylinders, it is customary to keep the air in the vaporizer at a pressure of several atmospheres, which greatly increases the capacity of the machine without increasing its dimensions.

Another type of air-refrigerating machine operates upon the regenerator principle. The air coming from the vaporizer is passed through a regenerator, where its temperature is increased almost to the temperature of the condenser. Its temperature is then raised still further by adiabatic compression and it enters the condenser, where it is cooled somewhat. It is then passed through the regenerator in the reverse direction, where it is cooled almost to the temperature of the vaporizer. It is then

expanded in a second cylinder and its temperature still further reduced before it is discharged into the vaporizer. It will be seen that in the case of such a machine, the amount of work performed upon the working fluid in the first cylinder, and by the working fluid in the second cylinder, is much less than in the case of the machine previously described, although the quantity of heat transferred per cycle by a given weight of working fluid is the same, when the temperature ranges of the two cycles are equal. In consequence of this fact, the regenerator cycle offers certain practical advantages in the matter of mechanical efficiency and cost of installation.

323. The Vapor-compression System. A vapor-compression machine is shown in principle in Fig. 181. Vapor (usually ammonia vapor) is compressed in the cylinder A, and then discharged at high pressure into the condenser B. The temperature of the cooling water being less than the saturation temperature of the vapor at the pressure in the con-

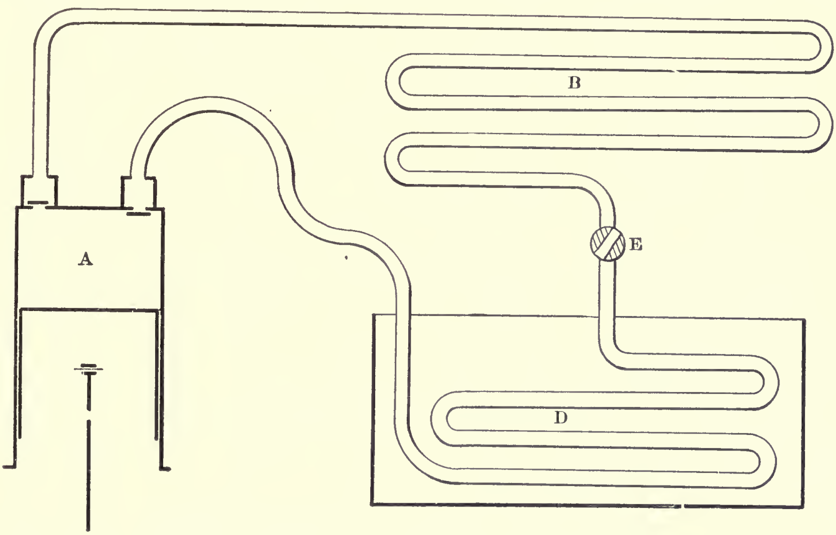


FIG. 181.—Ammonia-compression plant.

denser, the vapor gives up its heat of superheat and then its latent heat of evaporation, and so condenses to a liquid. After condensation, the liquid escapes through the expansion valve E into the vaporizer D, where it evaporates under low pressure by abstracting heat from its surroundings. The pump A draws the vapor from the cooling coils as fast as it is formed, and compresses it in order that it may repeat its cycle. Since the temperature of vaporization in the condenser is high, and since it is low in the vaporizer, on account of the low pressure, the machine is able to transfer heat from a cold region to a hot one.

It will be seen that no work is performed by the working fluid, so that it is evident that this cycle cannot be a very efficient one. It has the advantage, however, of being extremely convenient and of requir-

ing only a small cylinder, whose mechanical efficiency will be comparatively high. It is found in practice that the commercial efficiency of this type of machine is superior to the commercial efficiency of the air machine operating on the reversed Joule cycle.

324. Example of a Vapor Compression Cycle. In order to illustrate the action of this cycle, we may take the following example. The working fluid is assumed to be 1 pound of ammonia. The temperature range desired is the same as in Art. 322, and we will assume, as before, that the fluid must be worked between the temperature limits of 100° F. and −20° F. In order to solve the problem, it will be necessary to make use of a table to the properties of the vapor of ammonia which may be found in Peabody's tables. The pressure of ammonia having a temperature of −20 F. is 17.7 pounds absolute, which will be the pressure of the vapor in the vaporizer. In order to raise its temperature to 100°, the vapor must be compressed to a pressure of 210.7 pounds absolute, which will be the pressure of the ammonia in the condenser. The latent heat of evaporation of ammonia at a temperature of 100° F. is 486 B.T.U., which will be approximately the quantity of heat absorbed in condensing 1 pound of ammonia in the condenser. The heat of the liquid at 100° F. is 75 B.T.U. and at −20° F. it is −57 B.T.U. (i.e., 57 B.T.U. will be required in order to raise its temperature from −20° F to 32° F.) The latent heat of evaporation of ammonia at −20° F. is 582 B.T.U. Of this 75+57 =132 B.T.U. are supplied by the heat of the liquid of the ammonia, and the remainder, or 450 B.T.U., is absorbed by the vaporizer from the region which is to be cooled. The specific volume of ammonia vapor at −20° F., is 15. 2 cubic feet, and at 100° F. is 1.52 cubic feet. The ratio of compression is therefore 10, and the work done, if the compression is assumed to be hyperbolic is,

$$210.7 \times 144 \times 1.52 \times \log_e 10 = 106{,}000 \text{ ft.-lbs.},$$

which is the mechanical equivalent of 136.3 B.T.U. The heat transferred is, of course, the heat absorbed by the ammonia in the vaporizer and is 450 B.T.U. The efficiency of the apparatus is then

$$450 \div 136.3 = 330 \text{ per cent.}$$

In theory the compression of the ammonia is not hyperbolic. The working fluid performs a reversed Rankine cycle in the compressor and the amount of work done can be computed exactly by taking the difference between the total heat of 1 pound of ammonia vapor at a temperature of 100° and 1 pound at a temperature of −20° F. The quality (or superheat) of the vapor at 100° may be determined from its entropy, which is the same as its entropy at −20. The method of computing the work performed in the case of this cycle will thus be seen to be identical

with the method employed in computing the work done during a Rankine cycle, as described in Art. 152. This method is, however, much more tedious and probably not very much more accurate than the assumption of hyperbolic compression.

It will be seen from the figures given that the efficiency of the vapor-compression machine is slightly less than that of a machine operating on the reversed Joule cycle, but in practice, it is found that the mechanical efficiency and the capacity for a given size of cylinder is so much greater in the vapor-compression machine that the actual efficiency of the apparatus is much greater than that of the air-refrigerating machine.

325. The Vapor-absorption System. The principle of the absorption machine will be understood by reference to Fig. 182. *A* is a closed cylinder, termed the generator, partially filled with a solution of ammonia

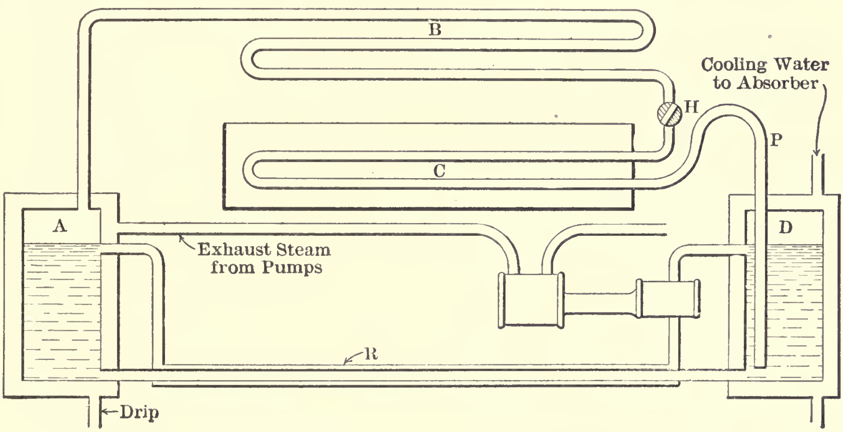


FIG. 182.—Ammonia-absorption plant.

gas in water under high pressure. On heating the generator by a fire or by steam coils, the ammonia is driven off through the pipe shown, into the condenser *B*. Since the ammonia is under high pressure, it is there condensed to a liquid when cooled by the condensing water. The liquid ammonia comes from the condenser at a temperature of perhaps 80° F., and passing through the expansion valve *H*, flows into the vaporizer *C*, where it evaporates. From the vaporizer the ammonia vapor passes through the pipe *p* into a vessel *d*, which is termed the absorber, and which contains a solution of ammonia. Brine is caused to circulate about the vaporizer coils and is then used to cool the region whose temperature it is desired to lower. The solution of the ammonia by the water in the absorber generates heat which is carried off by circulating cooling water through coils immersed in the absorber. The liquid contained in the absorber is removed by a pump and transferred to the generator through

the regenerator coil R. The spent liquid from the generator is transferred to the absorber, passing through this same generator coil on the way. The liquid entering the regenerator is thus heated while that entering the absorber is cooled.

It will be noted that the only power required by the absorption system is that required to circulate the various liquids and to force the liquid from the absorber into the generator. It will be seen that the power required is inconsiderable as compared with that required by the compression system. The amount of heat required by the generator is, of course, somewhat greater than the amount of heat required to vaporize the ammonia, so that at first sight it would appear that this system must have an efficiency of less than 100 per cent, which is very low for a refrigeration system. However, the efficiency of the engine which furnishes mechanical power for the compressor in the vapor-compression system is rarely greater than 10 per cent, so that the efficiency of the absorption system, when considered from the standpoint of the cost of operation and not of the quantity of energy required to effect the heat transfer, is very much greater than that of the compression system. The steam exhausted by the pumps used in connection with the vapor-absorption system usually furnishes sufficient heat to operate the system. The amount of cooling water taken by the system is much greater than that taken by a vapor-compression plant of the same capacity.

326. Apparatus for Liquefying Gases. When very low temperature is desired, as, for instance, when it is desired to liquefy any of the permanent gases, advantage is taken of the cooling which accompanies the expansion of the gas due to the work required to separate its particles against their mutual attractions. The apparatus which is employed to liquefy air is illustrated in principle in Fig. 183. It usually consists of 3- or 4-stage compressor which raises the pressure of the air to from 2000 to 2500 pounds per square inch. Next, this air is cooled in the coil C to the lowest available temperature, usually to the temperature of the coldest water available. In case the Joule-Thompson effect of the gas to be liquefied is small at ordinary temperatures, refrigerating agents may be employed to cool the gas in this coil. After being cooled, it is allowed to flow through a long tube which is enclosed within a second tube. At the end of this tube, the air passes through the reducing valve V, and expands into the flask F, in which it is proposed to collect the liquefied air. As a result of the expansion, the temperature of the air is lowered a few degrees. This air returns from the flask to the compressor through the outside tube. These two tubes act as a regenerator, and the temperature of the air coming to the flask through the inner tube is reduced by transferring its heat to the cooler air returning to the compressor through the outer tube. Since the temperature of the air entering the reducing valve is

lowered by the action of the regenerator, the temperature of the air coming from the reducing valve is lowered still further, and the action is cumulative, the temperature of the air being gradually reduced until finally it becomes low enough so that a portion of it is liquefied. After passing through the expansion valve, a portion of the liquid remains behind in the flask. As soon as liquefaction commences, no further reduction in temperature takes place, but the air continues to liquefy, and the fresh air, which must be dried and freed from carbon dioxide in order to avoid difficulties from the formation of ice or carbon dioxide snow in the apparatus, must be taken into the compressor to take the place of the air which is liquefied.

It will be seen that the heat which is extracted from the air in cooling it after compression is greater than the work of adiabatic compression by the amount of work done by the attraction of the particles of the air

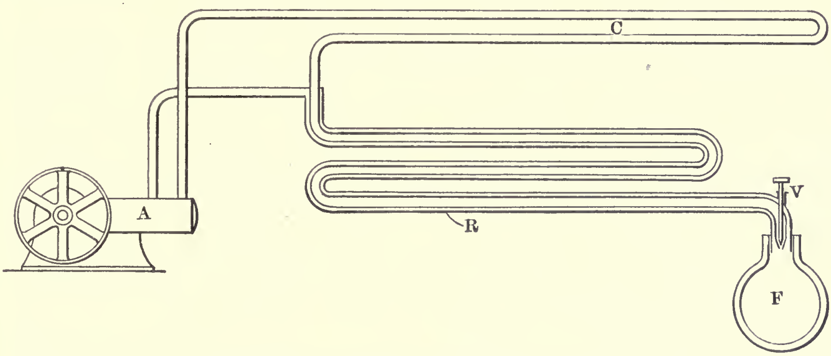


FIG. 183.—Apparatus for liquefying air.

while they were being forced together; this heat was transferred to the cooling water from the air which is liquefied, by the action of the regenerator.

This type of apparatus is expensive and not very efficient, but may be employed to advantage when very low temperatures are needed for scientific investigations. Most of the known gases have been liquefied by the employment of this apparatus, and there is no reason to believe that there are any gases which cannot be liquefied in this manner when they can be obtained in sufficient quantities.

327. Conventional Methods of Stating Capacity and Efficiency. It is customary to rate refrigerating machines by the "ice-melting effect" in tons per twenty-four hours. The quantity of heat required for the fusion of 1 pound of ice is very nearly 142 B.T.U. Consequently the quantity of heat absorbed by the fusion of 1 ton of ice is $142 \times 2000 =$ 284,000 B.T.U. A 1-ton refrigerating machine or system is then a

machine or system which is capable of removing 284,000 B T U. per day of twenty-four hours from the vaporizer. Such a machine will, in a commercial plant, usually be capable of freezing about 1000 pounds of ice per day. In comparing efficiencies of refrigerating machines, it is usual to state the ice-melting effect in pounds in per pound of coal or per indicated horse-power per hour, the indicated horse-power being the horse-power of the engine which drives the compressor. Since the quantity of heat transferred by a given expenditure of power will vary with the temperature range, being less for large temperature ranges than for small ones, it is customary to state the efficiency for a temperature range from 0° F. in the vaporizer to 90° F. in the condenser.

PROBLEMS

1. A refrigerating machine is required to maintain a temperature of 30° F. in a region in which condensing water is available having a temperature of 70° F. What is the maximum theoretical efficiency possible assuming a Carnot cycle to be employed? Ans. 1225%.

2. An air-refrigerating machine of the type described in Art. 313 is required to operate between the temperature limits given in Prob. 1. Assume that the air must be cooled by expansion to 10° F. and heated by compression to 90° F. If the pressure of the air in the vaporizer is 50 lbs. per square inch absolute, what must be the pressure of the air in the condenser? Ans. 85.7 lbs. per square inch.

3. Assume that the air is warmed 10° in the vaporizer. How much heat is transferred per pound of air per cycle? Ans. 2.37 B.T.U.

4. Find the swept volume of the large cylinder per pound of air per cycle, assuming that its volumetric efficiency is 80%. Ans. 4.35 cu.ft.

5. A machine making 60 revolutions per minute is required to transfer 7200 B.T.U. per hour from the vaporizer. What must be the swept volume of the large cylinder? Ans. 1.84 cu.ft.

6. An ammonia compression machine is required to maintain the temperature difference given in Prob. 1. Assume that the temperature in the vaporizer is 10°F., and the temperature in the condenser is 90° F. The pressure of ammonia at 10° is 37.8 lbs. per square inch, its specific volume is 7.44 cu.ft, the heat of the liquid is -24 B.T.U., the heat of vaporization is 558 B.T.U., and the entropy of the liquid is -0.0501, and the entropy of vaporization is $\frac{558}{469.6}$. What is the total heat of the vapor at this temperature? Ans. 534 B.T.U.

7. What is the entropy of the vapor after compression? Ans. 1.139

8. The pressure of ammonia at 90° is 179.6, the heat of the liquid is 64 B.T.U., the heat of vaporization is 494 B.T.U., the entropy of the liquid is 0.1224, the entropy of vaporiaztion is $\frac{494}{549.6}$, and the specific volume is 1.76. What is the entropy of ammonia vapor at 90°? Ans. 1.022.

9. Is the ammonia wet or superheated after compression? Ans. Superheated.

10. What quantity of heat is transferred from the vaporizer to the condenser per pound of ammonia per cycle. Ans. 470 B.T.U.

11. Assume that a compressor making 30 revolutions per minute is required to transfer 1,000,000 B.T.U. per hour from the vaporizer to the condenser? What quantity of ammonia must be compressed per revolution? Ans. 1.18 lbs.

12. Assuming that the volumetric efficiency of the compressor is 80 per cent, what swept volume per revolution will be required? Ans. 10.96 cu.ft.

13. Assuming hyperbolic compression, what work will be required per revolution to compress this quantity of ammonia? Ans. 75900 ft.-lbs.

14. What will be the horse-power required to drive the compressor if its mechanical efficiency is 70 per cent? Ans. 98.7 H.P.

15. What is the nominal capacity of the compressor in Problem 11?

Ans. 84.5 tons.

16. What is the efficiency of the compressor expressed in pounds of ice-melting effect per indicated horse-power per hour? Ans. 71.7 lbs.

17. Assuming a coal consumption of 3 lbs. per indicated horse-power per hour, what is the efficiency expressed in ice-melting effect per pound of coal?

Ans. 23.9 lbs.

CHAPTER XXIV

HEATING, VENTILATION, EVAPORATION, AND DRYING

328. The Hygiene of Heating and Ventilation. Within the human body a process of oxidation is continually going on. The products of oxidation are excreted by the lungs and the skin, and thrown off into the air. These products consist of carbon dioxide and water vapor, together with other vapors or gases of very small amount and unknown characteristics. Carbon dioxide was formerly thought to be poisonous, but it is now known that when the air contains less than 1 or 2 per cent of it, it has no effect upon animal life, being as inert as so much nitrogen. Water vapor is also harmless. We know, however, both as a result of scientific investigation and practical experience, that the exhalations from the human body are dangerous to life, and many authorities are of the opinion that the poisonous exhalations are thrown off by the skin rather than by the lungs. It has been demonstrated experimentally that when the quantity of animal exhalation present in air is great enough so that the carbon dioxide content of the air exceeds 0.07 per cent, the air is unfit to breathe. The carbon dioxide is not to be regarded as an objectionable component, but simply as an indicator which shows the suitability of the air for breathing.

The average adult requires about 20 cubic feet of air per hour for respiration, and exhales about 0.6 cubic feet per hour of carbon dioxide. Since it is impossible to avoid the mingling of the exhaled air and the fresh air supplied by ventilation, it is necessary to furnish very much more air per person than the 20 cubic feet actually consumed. If it be assumed that the air exhaled from the lungs mingles freely with the air supplied by ventilation, it is necessary to supply about 2000 cubic feet of fresh air per hour for each person present, in order to prevent the carbon content from rising above 0.07 per cent. In old treatises on ventilation it was assumed that the carbon dioxide was the dangerous constituent of the air, and hence that an additional supply of air was necessary in rooms containing open flames, such for instance as gas jets. Since such flames often give off carbon monoxide, sulphur dioxide, and other objectionable gases, it is advisable to provide extra ventilation in such a case, but this extra ventilation is not made necessary by the carbon dioxide. In large rooms containing a considerable volume of air per person and which are

used for short periods only, as for instance, churches and public halls, it is not necessary to supply 2000 cubic feet of air per person per hour, since some time will elapse before the air in the room is sufficiently vitiated to require renewal. On the other hand, in hospitals, particularly in contagious wards, it is advisable to supply a much larger quantity of air per person. In the case of dwelling houses, with unpainted plastered walls and in the case of many other forms of construction a considerable amount of ventilation is secured by diffusion through the walls of the rooms. When the composition of the air in a room having porous walls becomes appreciably different from that of the external air, diffusion takes place, which tends to make the composition of the air in the room identical with that of the external air. While this action may be relied upon to some extent to supply ventilation, it is not a satisfactory substitute for the movement of air in the form of a stream or current.

Not only is it necessary to supply an adequate amount of fresh air to effect the removal of the organic exhalations in any inhabited room, but is also necessary to keep the room at a proper temperature and the air in the room at a suitable humidity. The usual temperature at which living-rooms are maintained in America is 70° F. In Europe it is usual to maintain living-rooms at a temperature of about 60° F. Experiments in the so-called open-air schools and cold-air schools indicate, however, that in the case of children wearing ordinary winter house clothing and permitted a reasonable degree of activity, that a temperature between 40° and 50° is the most satisfactory room temperature. The reason for this is that the exhalations from the body, on account of the relatively high temperature, are then sufficiently lighter than the air in the room, so that they rise promptly from the breathing zone and pass out of the room without vitiating the air which the inhabitants are to breathe. Older persons, when engaged in sedentary occupations and especially those who have been accustomed to warm living-rooms, do not find such temperatures agreeable, however, and since it is usually the older persons who determine such matters, the temperature of living-rooms is usually maintained at a higher point than proper hygiene dictates. In most schools at the present time, 68° F. is prescribed as the proper maximum of temperature, and the tendency is to lower this maximum rather than to raise it.

The humidity of the air supplied by ventilation is quite as important a matter as is its temperature. The normal humidity of out-door air is about 70 per cent, and this degree of humidity in connection with a temperature between 60° and 70° F. seems to be most favorable to the proper performance of all the vital processes.[1]

[1] This is true only in case the ventilation is unusually abundant and effective. When it is not, a lower temperature is desirable in order that the convection currents

In an artificially heated building it is difficult to maintain the humidity of the air at a proper point, since the air is taken into the building at low temperature, and therefore contains but a small quantity of moisture, and its temperature is subsequently raised without increasing the moisture content. This makes the air exceedingly dry. The effect of such dry air upon the human body is, of course, to take moisture from the skin and mucous membranes very rapidly. This has a tendency to make the body feel cold on account of the rapid evaporation, to produce diseases of the nose and throat, and to seriously disturb the circulatory system. Hence if proper ventilation is to be maintained in a building, it is necessary to introduce steam into the air which is to be circulated by the ventilating system, in order that its humidity may be that which is proper for health.

329. Systems of Heating. Two systems of heating are employed, which are known as the direct and the indirect systems of heating. Direct-heating apparatus is apparatus which is placed in the room to be warmed. Stoves and radiators are apparatus of this type. Indirect heating apparatus is apparatus which is employed to heat a current of air which is then introduced into the room to be warmed. Direct-heating apparatus is employed principally in connection with dwelling houses, office buildings and other places where ventilation is not a matter of primary importance. Indirect-heating apparatus is employed in schools, hospitals and other places where adequate ventilation is of great importance.

330. Direct-heating Systems. The common coal stove is the simplest form of direct heating apparatus. It is reasonably efficient, but on account of the constant attention necessary, and the dirt created by the use of coal and the disposal of ash, the stove is being displaced by other forms of heating apparatus in which the fire is maintained in a place where the handling of dirt and ash is not objectionable.

The most common system of direct heating is that which employs steam as the medium for distributing the heat. Steam radiators consist of coils of pipe or of shells of cast iron or pressed steel which are supplied with steam by piping from a central boiler plant. The steam condenses within the radiator, transferring its heat to the air in contact with the shell. This heated air rises and cold air flows in to take its place, thus warming the air in the room to be heated. The condensation returns to the boiler through the piping system. The principal differences between the several systems of steam heating usually employed lie in the method of returning the condensation to the boiler. The simplest system is that illustrated in Fig. 184, amd is known as the single-pipe system. In this system a current of steam ascends from the boiler *B* through the

created by the heat of the body shall clear the breathing zone of undesirable exhalations.

pipe P to the radiator R, and the condensation returns through this same pipe to the boiler. It will be seen that it is necessary to make the pipe of ample size so that the current of steam will not have a high velocity, for if it has, it will retard the returning current of water and may cause the system to become "water bound." It is also necessary that the pipe shall be so arranged that every part of it shall drain freely into the boiler, for if a "pocket" is formed in the pipe which can fill with water, the system will become water bound, and the surging of this water through the pipes under the action of the steam will produce severe "water hammer."

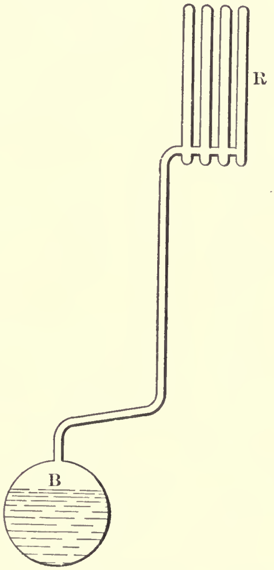


Fig. 184.

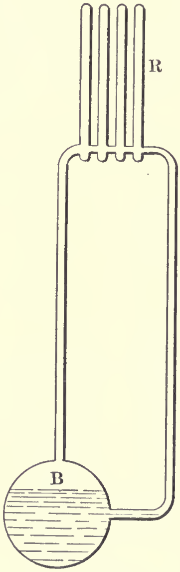


Fig. 185.

The return-pipe system, illustrated in Fig. 185, provides a separate pipe for the return of the condensation. With this system, the pipes may be made smaller than with the single-pipe system, and pockets, although they are to be avoided as far as possible, are not fatal to successful operation.

When a radiator system is started up the pipes and radiators are of course full of air. In order that steam may fill the system, it is necessary that the air be permitted to escape. The air usually escapes from the radiators through valves termed air valves, which are sometimes automatic in their action, but which are usually of a form requiring personal attention. If only a part of the air escapes from the radiator, those sections of the radiator which are filled with air do not permit the entrance of

steam, so that only a portion of the radiator will be active. This is desirable in mild weather, when only a small amount of heat is needed. The principal objection to a system of steam radiation is that unless intelligent advantage is taken of the effect of the presence of air in the radiator, it is necessary to leave the steam fully on or to shut it completely off. In order to avoid this difficulty radiators of the form shown in Fig. 186 are sometimes used. In these radiators, the steam enters at the top, and the condensation flows away at the bottom. The steam enters the radiator through the throttle valve V, which may be opened sufficiently to admit the desired quantity of steam. The air, being heavier than the steam, is forced out through the return pipe which is connected into the bottom of the radiator. A balance is quickly established between the quantity of steam supplied and of steam condensed, so that air occupies the bottom of the sections and steam the top, and the amount of heat radiated is determined by the amount of the radiating surface in contact with steam.

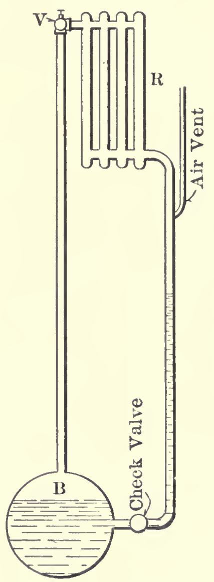


Fig. 186.

The amount of radiating surface required when steam radiation is employed is usually determined on the assumption that 250 B.T.U. are transferred per hour from the steam to the air by each square foot of radiating surface. A sufficient amount of surface is provided on this assumption, so that the amount of heat given up by the radiators to the air in the room is equal to the amount of heat lost by the room through radiation and ventilation.

In order to estimate the amount of heat lost by a room, it is customary to employ the formula

$$H = \left(c\left(G + \frac{W}{4}\right) + \frac{nC}{56}\right)(T_1 - T_0).$$

In this equation H is the number of heat units required per hour for heating and ventilating the given room, c is a factor varying from 1.1 to 1.3 and depending on the exposure of the room, the direction of the prevailing winds etc., G is the number of square feet of glass surface in the windows, W is the number of square feet of wall surface exposed to outdoor air, n is the number of air changes required per hour for ventilation, C is the number of cubic feet of air space in the room, T_1 is the temperature at which the room is to be maintained, and T_0 is the lowest outdoor temperature which it is desirable to provide against.

The following example will serve to show the method of computing the amount of radiating surface which will be required in a given room. Assume a room 30 feet long, 20 feet wide, and 15 feet high having a side and an end wall exposed to the external air. Assume that the room is provided with four windows, which are each 4 feet wide and 10 feet high, and that it is to house 40 persons. The number of cubic feet of air required per hour will be $40\times 2000\times 80{,}000$, which is the value of $n\ C$ in the formula. The value of G in the formula will be $4\times 4\times 10=160$ square feet of window surface. The value of W, the exposed wall surface, will be $(15\times 20+15\times 30)-160=590$ square feet. The value of c we will assume to be 1.2 and we will also assume that the room is to be maintained at the temperature of 70° when the temperature of the external air is zero. We will then have for the number of heat units per hour for heating and ventilating the room

$$H=\left(1.2\left(160+\frac{590}{4}\right)+\frac{80{,}000}{56}\right)70=126{,}000.$$

For the number of square feet of radiating surface needed we will have

$$\frac{126{,}000}{250}=504.$$

331. Exhaust-steam Heating. When exhaust steam from an engine is available, it may be used for heating. In office buildings it is customary to install a plant in which steam engines are used for furnishing light and power for the building and the exhaust steam is employed for heating. The engine usually exhausts into a tank or header, to which the piping system is connected, as shown in Fig. 187. The header is provided with a relief valve R, whose purpose it is to allow the escape of steam when its pressure exceeds a certain value. The steam passes from the header to the heating system, and so long as the amount of steam supplied by the engine is great enough, the excess will escape through the relief valve. When the amount of steam supplied by the engine is not sufficient to heat the building, steam enters the header from the boiler through the pressure-reducing valve V. If the relief valve

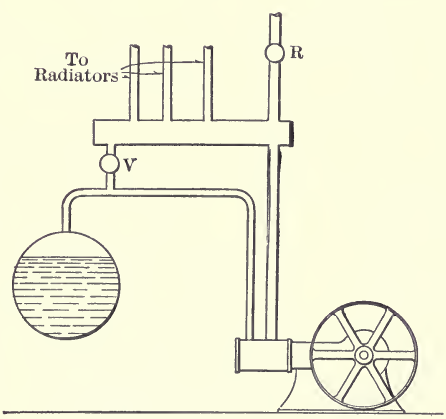


Fig. 187.

be set to operate at 3 pounds gage, while the pressure-reducing valve is set to operate at 2 pounds gage, it will be seen that the tank will always contain steam at a pressure of between 2 and 3 pounds.

In some systems, termed vacuum-heating systems, the air is removed from the system by means of a vacuum pump or a steam jet and the pressure of the steam within the system is maintained at some value less than atmospheric pressure. The advantage of this system of operation is that it reduces the quantity of steam required by the engine. It is especially adapted to those cases where the amount of steam exhausted by the engine, when operated under high back pressure, would be greater than the amount of steam required for heating purposes. With this type of heating plant, it is important that the system be made air-tight by carefully making all joints and packing the valves so that the quantity of air to be handled by the vacuum pump will be a minimum.

332. Hot-water Heating. A system of direct radiation often used in domestic heating is known as hot-water heating. In this system water is heated in an apparatus similar to a boiler, termed a heater. The water is caused to circulate through radiators, and after being cooled, is returned to the heater. The circulation is produced by the difference in the density of the hot water coming from the heater and the cold water which has passed through the radiators and is returning to the heater. A hot water plant is shown in principle in Fig. 188. *H* is the heater, which is located at the lowest point in the system and *R* is the riser which supplies the radiators with hot water. This riser terminates in a tank, *E*, termed the expansion tank. The purpose of this tank is to permit the expansion of the water without allowing it to escape from the system. Since the water in the riser *R* will, on account of its temperature, be lighter than the water in the return pipe *P*, it will flow through the radiators, surrendering its heat to the air in contact with them. By partially closing the valve which supplies the water to the radiator, the amount of heat radiated may be controlled, which is a great advantage in house-heating in mild weather. Since the temperature of the water in the radiator averages much lower than the temperature of the steam ordinarily supplied to radiators, it will be seen that a large radiating surface is necessary. It is usual to assume that 1 square foot of hot water radiation will supply 180 B.T.U. per hour to the room in which it is situated.

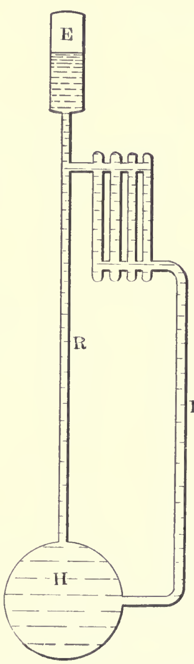


FIG. 188.

333. Indirect Heating. Two systems of indirect heating are in use. In the first system the difference in density between the hot air in the ventilating flues and in the cold air in the building, is depended upon in order to circulate the air required for ventilation; in the second system a fan or other mechanical impeller forces the air to the proper point. The hot-air furnace employed in heating dwelling houses is an example of the first method. Such a heating system is illustrated in principle in Fig. 189. The hot-air furnace is placed at some point lower than the lowest point which it is required to heat. The air is warmed by a fire which is separated from it by a partition, usually of cast iron. The air is carried by flues to the rooms to be warmed. Since the warm air in the flues is lighter than the air in the rooms, the air in the rooms descends to the basement, where it enters the furnace and rises through the flues and registers. When the furnace draws its supply of air to be warmed from the basement itself, the air in the house is not renewed. If, however, the furnace draws a part or all of its supply of air from out of doors through a "cold air box," and the air in the house is allowed to escape instead of being returned to the basement, ventilation is secured. In public buildings where large numbers of people congregate, it is undesirable to recirculate the air, since the demands of ventilation are such that, if the air were used over again, it would be too foul. In the case of dwelling houses, however, on account of the small number of inhabitants as compared with the radiating surface of the house, it is desirable to recirculate a large part of the air used, in order to conserve heat.

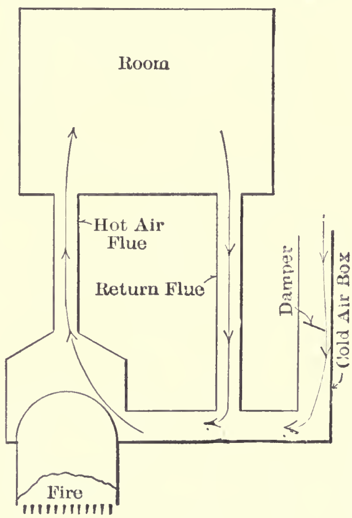


Fig. 189.

When large buildings are to be heated and ventilated, indirect steam heating is often employed. In this system, steam coils or radiators of special form are placed in the ventilating flues, at a point below the level of the floor of the room to be warmed. Since the air in the flue is highly heated by the radiator, it rises into the room, forcing out the cold foul air which the room contains. On account of the vigorous air circulation and the form of radiator usually employed, a square foot of indirect radiating surface will impart 400 B.T.U. per hour to the air passing it.

334. Forced Ventilation. In the ventilating systems previously described, dependence was placed upon the difference in temperature

between the air in the ventilating flues and the air in the building in order to secure a proper circulation. However, since weather conditions will often destroy the effectiveness of such a system, it is advisable where adequate ventilation is essential under all weather conditions, to install some mechanical device, such as a fan, for moving the air, in order to insure that a sufficient quantity of it shall be moved to the places where it is needed. Such a system of mechanical ventilation is illustrated in Fig. 190. A boiler *B* supplies steam to coils of pipe placed in the ventilating duct *D*. A fan *F* supplies the air used for ventilation. Through the jets *J* the steam is introduced to properly humidify this air after it has

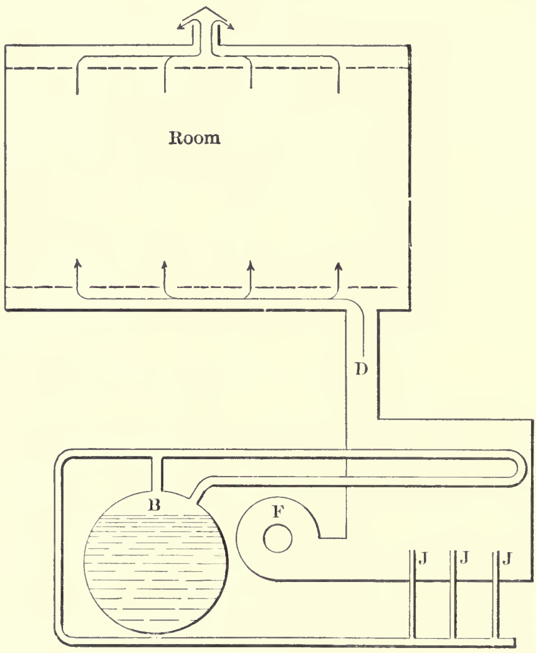


FIG. 190.

been warmed. The air rises through the duct to the room which is to be heated. The best method of distributing the air in the room is still an open question. It would be natural to introduce the warm air at the top of the room and to allow it to expel the cool air at the bottom of the room. Since, however, the exhalations from the body are warmer than the air contained in the room, they would tend to rise and then to be forced down again into the breathing zone by the down-coming and slow-moving current of air from above. The best method, (which, however, is quite expensive in application) is to admit the air at the bottom of the room in the manner shown, so that it is uniformly distributed to all parts of the room. The foul air is removed at the top of the room. If incom-

ing air were supplied at localized points, as it would be if supplied through registers, the fresh air would immediately rise to the top of the building and there escape, leaving the foul air behind. By distributing it through very numerous small openings, spaced uniformly over the entire floor area, this difficulty may be avoided.

In some cases it is desirable that the air introduced into a building shall be free from dust. This is especially the case in hospitals, since dust acts as a germ carrier.[1] The removal of dust is accomplished by passing the air through a scrubber or other form of gas-washing apparatus before the air is heated.

335. Evaporation. In many of the chemical industries it is necessary to evaporate solutions of salts in order to obtain the salts in solid form. In other industries it is often necessary to concentrate solutions by evaporating the larger part of the liquid which they contain. The

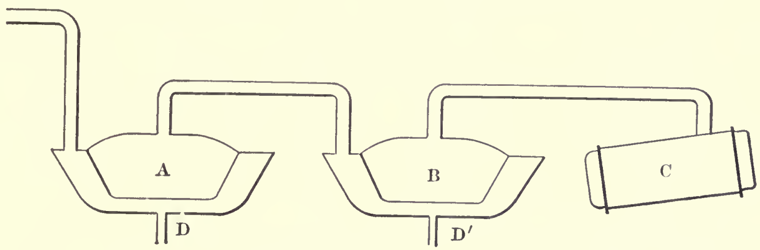


Fig. 191.—Diagram of a double effect evaporator.

preparation of salt or sugar are examples of such industries. In order to evaporate the water contained, it is usually necessary to heat the solution to a temperature considerably higher than the saturation temperature of the steam coming from the solution. In order to evaporate a maximum quantity of water by the use of a given quantity of heat (i.e., in order to make the evaporator as economical as possible), it is customary to make use of an apparatus usually termed a **double-** or **triple-effect evaporator.** The principle of operation of a double-effect evaporator may be seen by reference to Fig. 191. The evaporation of the liquid occurs in the pans A and B. These pans are provided with steam jackets. The jackets are shown surrounding the pans in the illustrations, although the heat is commonly applied by causing the steam to pass through coils immersed in the liquid. Steam is supplied to the jacket of pan A under high pressure, say 100 pounds, per square inch. The temperature of steam at that pressure is 328° F. The steam which is evaporated from the liquid contained in the pan is used to jacket pan B. Its pressure will be, let us say, 20 pounds per square inch absolute, which corresponds to a temperature

[1] So far as it is known, all air-borne germs are transported while adhering to particles of floating dust. The elimination of dust from the air supplied for ventilation effectually excludes such germs from a room.

of vaporization 228°. The difference in temperature of 100° F. is sufficient to evaporate the liquid contained in the pan A in spite of the fact that the temperature of the liquid is considerably higher than the 228° corresponding to the pressure of the steam formed. The steam which is evaporated from pan B is of a pressure, let us say, of 2 pounds absolute, or at a temperature of 126°. This steam is condensed by the condenser C. A vacuum pump is employed to remove the air which is brought into the system by the liquid which is to be evaporated. The condensation from the jackets flows away through the drips D and D' to traps which permit it to escape. It will be seen that by the employment of such an apparatus, the heat which would otherwise be rejected with the steam evaporated from the first pan is utilized in evaporating practically the same weight of liquid in the second pan. This results in doubling the efficiency of the apparatus. Where a high efficiency is desired, three or even more pans may be employed in series. Since the temperature differences between the steam in the jacket and the liquid in the pan will be reduced by increasing the number of pans, it follows that the first cost of the apparatus required to evaporate a given weight of liquid per hour will increase as the efficiency is raised by increasing the number of pans in series. Evaporators are of many forms, and are provided with numerous mechanical devices, in much the same way as are steam boilers, in order to make their operation more efficient and convenient.

When exhaust steam is available it is customary to employ it for heating the first pan in a double-effect evaporator, making the temperature drop per pan approximately 50°. The capacity of the system will be quite largely increased by improving the performance of the vacuum pumps and so arranging the system that the air to be handled by the pumps will be drawn from the coolest part of the system and will be mingled with the minimum quantity of vapor. Vacuum pumps are not needed for those parts of the system where the pressure of the steam is greater than that of the atmosphere, since a small portion of the steam may be permitted to escape through a valve, carrying the air with it.

336. Distilling. In some cases, it is the vapor which is evaporated and not the substance which remains behind in the pan, which is the valuable product. In such a case, the vapor must be condensed by the use of an apparatus termed a **still.** A still consists of an evaporating chamber heated by fire or by steam, and of a coil of pipe usually termed a worm, which is immersed in a tank supplied with cooling water at the bottom, the water being drawn away at the top. Any form of apparatus which will act as a condenser, however, is equally suitable for use as a still. Stills are employed in the preparation of alcohol, liquid ammonia, kerosene and many other volatile liquids. When the liquids are likely to bring into a still uncondensible gases, such as air, it is desirable to provide

the still with a vacuum pump in order that the pressure and therefore the temperature of evaporation shall be as low as possible. Stills may be arranged so as to operate in series, the vapor condensed in one worm acting to evaporate the liquid in the second still.

337. Drying. The drying of substances containing only small quantities of moisture, as for instance, lumber cloth, etc., is usually effected by exposing the substances to be dried to a current of dry air. At atmospheric temperature air, of course, contains water vapor. Since the pressure of the water vapor is less than the saturation pressure corresponding to the temperature of the air, the air will absorb moisture. It may be caused to absorb moisture much more rapidly, however, by heating it, which will decrease its relative humidity and increase very greatly its capacity for absorbing moisture by raising the pressure of the water vapor which it can contain. If such a current of heated air be caused to pass through a pile of lumber, as is done in the kiln-drying process, it rapidly absorbs moisture from the lumber and passes out of the kiln with a much larger moisture content than it had on entering. The air is usually caused to circulate by a fan or other form of mechanical impeller and is heated by the use of coils of steam pipe. These coils are usually supplied with exhaust steam.

PROBLEMS

1. How many cubic feet of air per hour will be necessary to properly ventilate a schoolroom in which there are thirty persons? Ans. 60,000 cu.ft.

2. Assuming that this air is taken into the building at a temperature of 32° F. and a humidity of 70 per cent, how many pounds of moisture will if contain? Ans. 12.8 lbs.

3. How many pounds of steam must be introduced into this air if the humidity of the room is to be maintained at 70 per cent and the temperature at 68°? Ans. 47.6 lbs.

4. A room 30 ft. long, 20 ft. wide, and 10 ft. high is exposed at one side and both ends. It contains six windows each $3\frac{1}{2}' \times 6'$. What quantity of heat will be required per hour to provide against the radiation loss, if the room is to be maintained at a temperature of 68° F., when the outdoor temperature is 10° F., the exposure being severe? Ans. 20,300 B.T.U.

5. Assuming that four changes of air per hour will be required in Prob. 4, what total quantity of heat will be required per hour? Ans. 44,700 B.T.U.

6. What quantity of direct steam radiation will be required to heat the above room? Ans. 179 sq.ft.

7. What quantity of hot-water radiation will be required to heat the above room? Ans. 248 sq.ft.

8. How many cubic feet of air per hour must be supplied by the ventilating system to the above room? Ans. 24,000 cu.ft.

9. At what temperature must the air be introduced in order that when it is cooled down to 68°, it shall part with sufficient heat to provide against the loss of heat by radiation? (Assume that the water equivalent of 1 cu.ft. of air is 0.018.) Ans. 115°.

10. How many square feet of indirect radiation will be required to warm this air from a temperature of 10° F.? Ans. 117 sq.ft.

CHAPTER XXV

ENTROPY DIAGRAMS

338. Nature of the Temperature-entropy Diagram. Entropy was defined in Art. 64 as the sum of the successive increments of heat necessary to bring a body from a fixed state to any given state, each divided by the absolute temperature at which the increment of heat occurred, and the definition expressed mathematically by the equation

$$\Delta N = \frac{\Delta H}{T}, \quad \text{. . . . .} \quad (1)$$

in which ΔN is the change in entropy, ΔH is the heat added (or abstracted) to produce this change of entropy, and T is the absolute temperature at which the change occurs. This expression may be transformed into

$$\Delta H = T \Delta N, \quad \text{. . . . .} \quad (2)$$

It is very convenient and instructive in certain cases, to represent thermodynamic processes by plotting on a diagram the relation between the temperature and the entropy of the body undergoing the processes. Such a diagram is called a **temperature-entropy diagram.** Its ordinates are proportional to absolute temperatures, its abscissæ to entropy, and its areas, as may be seen from equation (2), are proportional to heat. Lines on such a diagram are termed temperature-entropy lines. Such a diagram may be plotted for any weight of substance, but it is usual to plot it for 1 pound of the substance.

339. Forms of the Temperature-entropy Lines. Assume a body whose mass is W, whose specific heat is S, whose temperature T_0, and whose entropy is zero (i.e., a body having the **zero state**), to have its temperature raised to the value T. The head added in order to effect a given small increase in temperature will of course be

$$dH = W\,S\,dT. \quad \text{. . . . .} \quad (1)$$

The corresponding change in entropy will be

$$dN = \frac{W\,S\,dT}{T}. \quad \text{. . . . .} \quad (2)$$

The entire change in entropy will be

$$\int_0^N dN = W\,S\int_{T_0}^{T}\frac{dT}{T}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Integrating, we will have

$$N = W\,S\log_e\frac{T}{T_0}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Plotting this relation on a temperature-entropy diagram, we will have the temperature-entropy line shown in Fig. 192. As may be seen from this figure, the entropy of the body at the zero state (i.e., the temperature T_0) is zero, the entropy of the body at temperature T is N, and the quan-

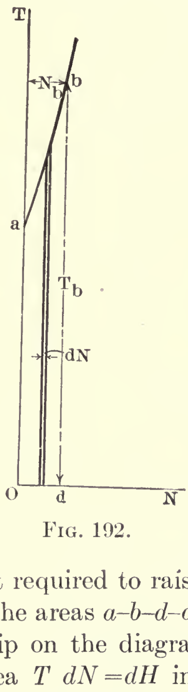


FIG. 192.

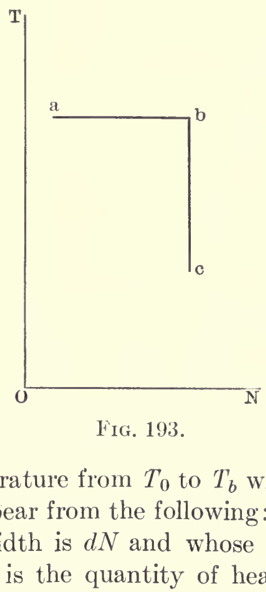


FIG. 193.

tity of heat required to raise the temperature from T_0 to T_b will be represented by the areas *a–b–d–o*, as will appear from the following:

The strip on the diagram whose width is dN and whose height is T has the area $T\,dN = dH$ in which dH is the quantity of heat imparted in order to change the entropy by the quantity dN. The total quantity of heat imparted in changing the body from the zero state to state B is the sum of all such strips included under the line *a–b*. Hence, the heat imparted is equal to the area included under the line.

When a body is heated without raising its temperature, as, for instance, when water is evaporated into steam, or a quantity of gas expands adiabatically, the increase of entropy occurs at constant temperature and the temperature-entropy line is horizontal, as, for instance, the line *a–b* in Fig. 193. When a body changes in temperature without receiving or

emitting heat, as for instance, when a quantity of gas or vapor expands adiabatically, the entropy is unchanged and the temperature-entropy line is vertical, as is the line b–c in Fig. 193. When a body absorbs or emits heat and simultaneously changes in temperature, and the quantity of heat absorbed or emitted is proportional to the changes in temperature, the temperature-entropy line will be of the form represented by the equation

$$N = K \log_e \frac{T_1}{T}. \qquad (5)$$

Such will be the case when the specific heat of the body is constant, or when a gas is heated at constant pressure or constant volume or undergoes polytropic expansion. When K is positive (i.e., when conditions

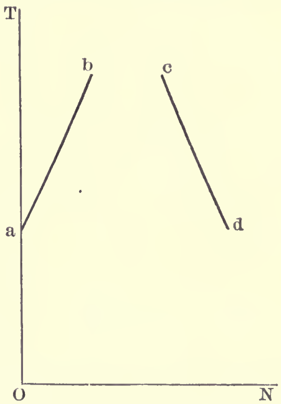


Fig. 194.

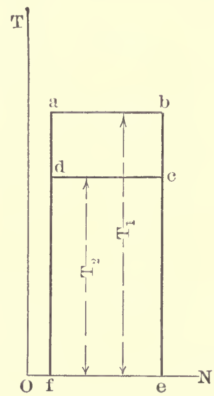


Fig. 195.

are such that the body rises in temperature as heat is added, or falls in temperature as heat is abstracted, the general direction of the temperature-entropy line will be like that of a-b in Fig. 194. When K is negative (i.e., when a fall in temperature accompanies an absorption of heat or a rise in temperature and emission of heat) the general direction is that of line $c\ d$ in the same figure.

340. Thermodynamic Cycles on the Temperature-entropy Plane. Any thermodynamic process involving a body of constant mass may be represented by an appropriate temperature-entropy line. Any series of such processes may be represented by a series of temperature-entropy lines, just as they may be represented by a series of pressure volume lines. Such a diagram may be drawn to represent any thermodynamic cycle, and the cycle is then said to be represented on the **temperature-entropy plane.** The diagram in Fig. 195 represents the Carnot cycle in which the working fluid expands isothermally from state a to state b at the

temperature T_1, receiving from the heater the quantity of heat represented by the area a–b–e–f. The line b–c represents the process of adiabatic expansion. The line c–d represents the process of isothermal compression at the temperature T_2, the quantity of heat represented by the area c–d–f–e being rejected to the cooler during this process. The line d–a represents the process of adiabatic compression. An inspection of this diagram will serve to show that the heat absorbed is to the heat rejected as the temperature T_1 is to the temperature T_2. Consequently, it may be shown from the diagram that the efficiency of the Carnot cycle, which is

$$\frac{H_1 - H_2}{H_1},$$

is also equal to

$$\frac{T_1 - T_2}{T_1}.$$

Any other cycle in which a constant mass of working fluid is employed or which is the equivalent of a cycle in which a constant mass of working fluid is employed, may be represented by the temperature-entropy diagram. In case, however, a variable quantity of working fluid is employed, as is the case in the practical steam-engine cycle, only those processes can be properly represented on a temperature-entropy diagram during which the quantity of working fluid remains constant.

341. Evaporation and Superheating on the Temperature-entropy Plane. The process of raising a quantity of water from the zero state (i.e., from the liquid state at a temperature of 32°F) to any temperature T, evaporating it into steam, and then superheating the steam at constant pressure, is represented on the temperature-entropy diagram by three lines of the form shown in Fig. 196. Line a–b represents the process of raising the temperature of the liquid from 32° to T. Since the specific heat of water is very nearly but not exactly a constant quantity, this line may be represented very nearly, but not exactly by the equation

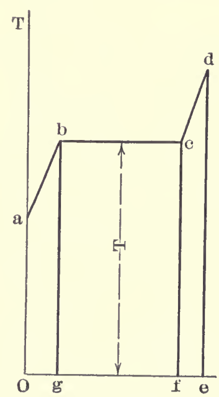


FIG. 196.

$$N = K \log_e \frac{T}{T_0}. \qquad (1)$$

During the evaporation of the water the temperature of the entire mass remains constant and the entropy increases. The process is the equivalent of isothermal expansion, and is represented by the line b–d. The process of superheating the steam is represented by the line c–d, which has approx-

imately the logarithmic form already given, although it does not approximate this form as nearly as does the line *a–b*, since the specific heat of steam at most pressures varies a little with the temperature. On this diagram, the area *a–b–g–o* represents the heat of the liquid, the area *b–c–f–g* represents the heat of evaporation, and the area *c–d–e–f* represents the heat of superheat.

342. The Steam Dome. Fig. 197 represents a form of temperature diagram often termed the **steam dome.** The line *a–b* represents the relation between the temperature and entropy of water, and is termed the

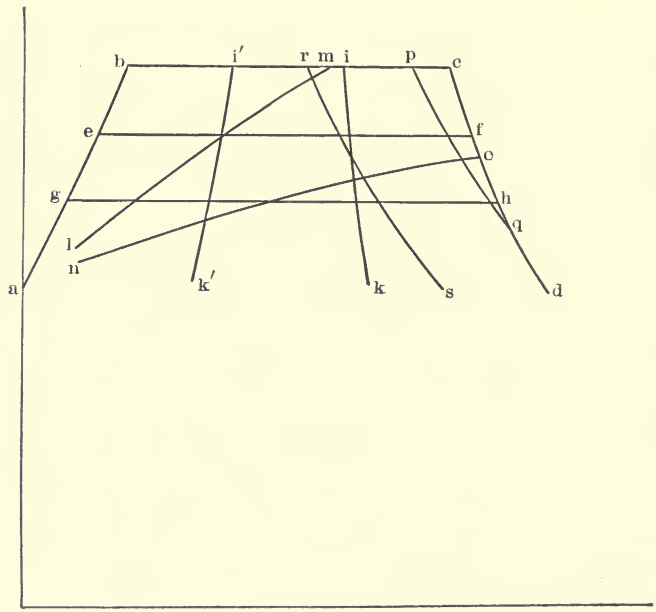


FIG. 197.

water line. The line *c–d* represents the relation between the temperature and entropy [1] of dry and saturated steam, and is termed the **saturation line.** Horizontal lines between *ab*, and *cd*, like *b–c*, *e–f*, *g–h*, etc., represent the relation between the temperature and entropy of wet steam of a given constant temperature, but of varying quality. The lengths of any of these lines, in entropy units, is, of course, the entropy of evaporation of dry and saturated steam of the given temperature. Since the quality of steam of a given pressure is proportional to its latent heat of evaporation, which in turn is proportional to its heat of evaporation, which in turn is proportional to its entropy of evaporation, any temperature-entropy line drawn in such a way as to divide these horizontal lines into

[1] When the entropy as steam is mentioned in this chapter, the total entropy is meant.

segments bearing a constant ratio to one another, is a **line of constant quality,** and represents the relation between the temperature and entropy of steam of a given quality. Line *i–k* is such a line, and the ratio of any segment, as *bi*, to the whole line, as *bc*, gives the quality of the steam. Vertical lines on the steam dome are, of course, lines of constant entropy. Lines of the form *l–m* and *n–o* are **lines of constant volume** and represent the relation between the temperature and entropy of a given volume of wet steam at different pressures. Lines of the form *p–q* and *r–s* are **lines of constant total heat,** and represents the relation between the temperature and entropy of wet steam of varying pressure but of a given total heat. In order to save space, it is customary to cut off that part of this diagram which lies below the temperature of 32° F., (i. e. the point *a*) unless it is necessary to use it for some reason.

An inspection of this diagram will serve to make clear a great many points in regard to the properties of steam. It will be seen, for instance, that dry steam expanding adiabatically becomes wet. On the other hand, steam having a quality of less than about 50 per cent becomes dryer as a result of adiabatic expansion. It will be seen that when steam expands without alteration in its total heat, as it does when throttled, the steam becomes dryer. Were the lines *a–b* and *c–d* continued upward sufficiently, they would finally meet at the critical temperature of the vapor.

343. Temperature-Entropy Diagram for Steam Cycles. The temperature-entropy diagram of the Carnot cycle for steam is exactly the same as it would be for any working fluid absorbing the same quantity of heat and working through the same temperature range. In Fig. 198 will be found the temperature-entropy diagrams of Carnot cycles for steam superimposed upon the steam dome, for different conditions, each diagram being accompanied by the corresponding pressure volume diagram.

The temperature-entropy diagram of the Rankine cycle for dry steam is illustrated in Fig. 199. Line *b–c* represents the isothermal expansion of the steam as it enters the cylinder, line *c–d* represents adiabatic expansion, line *d–e* represents the isothermal compression and condensation, and line *e-b* represents the heating of the condensing steam to its initial temperature. The work done is, of course, represented by the area *b–c–d–e*. The heat imparted is represented by the area *b–c–f–g–e*. An inspection of the diagram will show that the efficiency of the Rankine cycle must be less than that of the Carnot cycle. That portion of the heat supplied which is represented by the area *e–b–h–g*, performs work represented by the area *e–b–i*, and the efficiency of this portion of the heat supply is $\frac{ebi}{ebhg}$. This is manifestly less than the efficiency of the

Carnot cycle working through the same temperature range, which is

$$\frac{jbie}{jbhg}.$$

The temperature-entropy diagram of the Rankine cycle, using wet steam, is shown in Fig. 200. It will be seen from this figure that the proportion of the total heat used inefficiently becomes greater as the quality

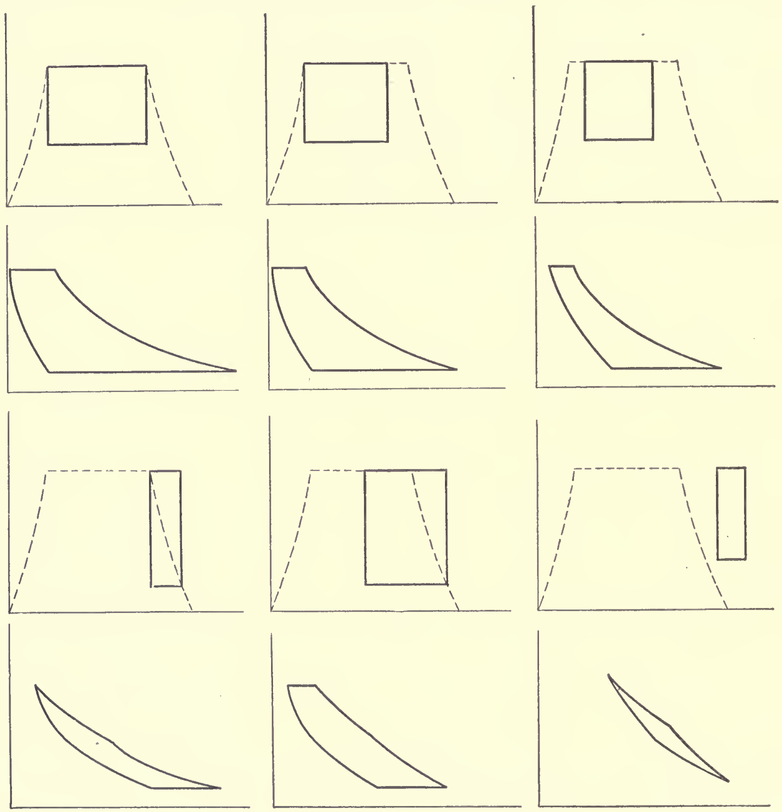


Fig. 198.

of the steam becomes less. Hence the efficiency of the cycle is less when wet steam is used. The diagram of the Rankine cycle using superheated steam is shown in Fig. 201, from which it may be seen that the superheated steam is more efficient than saturated steam of the same pressure, but not as efficient as saturated steam of the same temperature. It will be noted that, as the steam expands, its superheat decreases and finally at the point i, the adiabatic crosses the saturation line and the steam becomes wet.

A temperature-entropy diagram of the modified Rankine cycle may be seen in Fig. 202. The cylinder contains a pound of working fluid, a portion

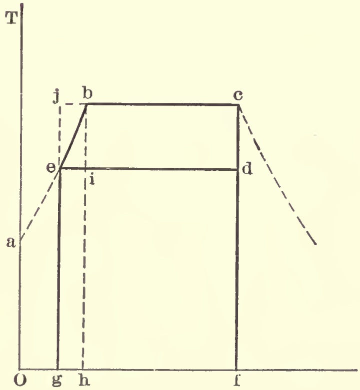


FIG. 199.

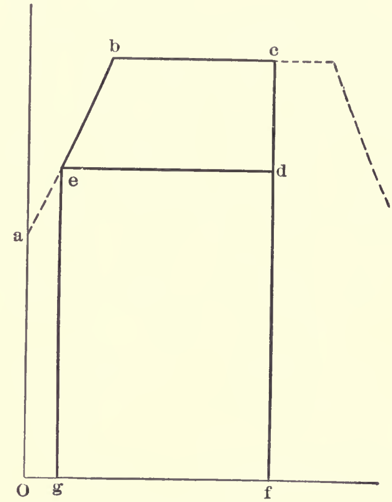


FIG. 200.

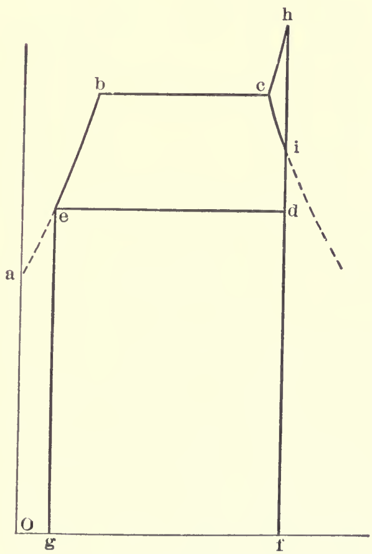


FIG. 201.

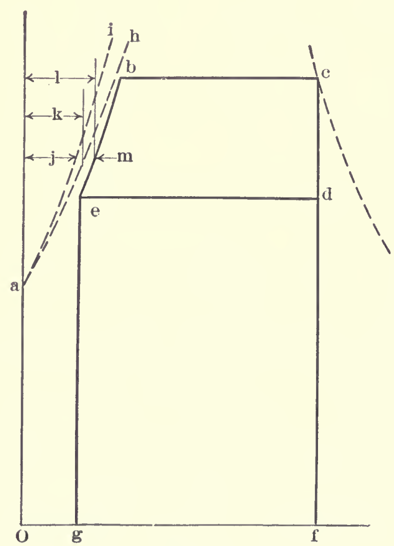


FIG. 202.

of which remains in the clearance space and is adiabatically compressed, while the remainder passes to the boiler where its temperature is raised while it is in the liquid state, by the application of heat. In order to

draw the temperature-entropy diagram of this cycle, it is necessary to assume that the water in the boiler has the same temperature at every instant during the compression period as does the cushion steam. The cushion steam is of course compressed adiabatically, but the compression line *e–b* is not an adiabatic, since it represents the relation between the temperature and entropy of the whole quantity of working fluid and not simply that of the cushion steam. In this diagram, line *a–h* is the water line for 1 pound of working fluid and line *a i* is the water line for that weight of working fluid rejected from the cylinder each cycle. The abscissa *j* represents the entropy of the liquid rejected from the cycle at some temperature, the abscissa *k* represents the entropy of 1 pound of water at the same temperature, the abscissa *l* represents the total

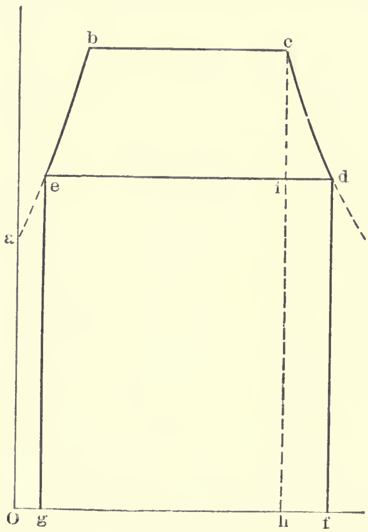


FIG. 203.

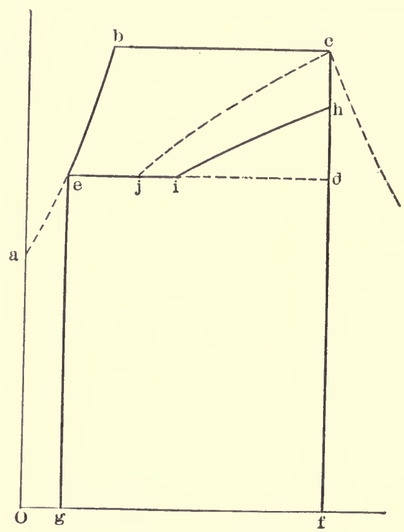


FIG. 204.

entropy of the working fluid during the compression period, at that temperature, and the distance m (which is equal to $l-j$) represents the total entropy of the cushion steam at that temperature. Since the cushion steam is compressed adiabatically, its entropy remains constant during the compression period, and the distance m also remains constant.

The temperature-entropy diagram of the jacketed cycle with complete expansion is shown in Fig. 203. The heat added is now represented by the area *e–b–c–d–f–g*, while the work performed is represented by the area *e–b–c–d*. The heat added by the jacket is represented by the area *c–d–f–h*, and the work done by this heat by the area *c–d–i*. It will be apparent that the heat supplied by the jacket is used less efficiently than that supplied by the cylinder feed, and that the efficiency of the jacketed cycle is theoretically less than that of the unjacketed cycle.

The temperature-entropy diagram of the imperfect cycle without clearance and using dry steam is shown in Fig. 204. That portion of the cycle lying within the area *i h d*, and cut off from the remainder of the diagram by the constant volume line *i–h*, is work lost on account of incomplete expansion. The less complete the expansion, the greater will be the quantity of work so lost, the constant-volume line *c–j* representing the limiting condition in which there is no expansion and the indicator card given by the engine is rectangular. The temperature-entropy diagram of the jacketed cycle with incomplete expansion is shown in Fig. 205, from which it may be seen that in case the expansion is incomplete,

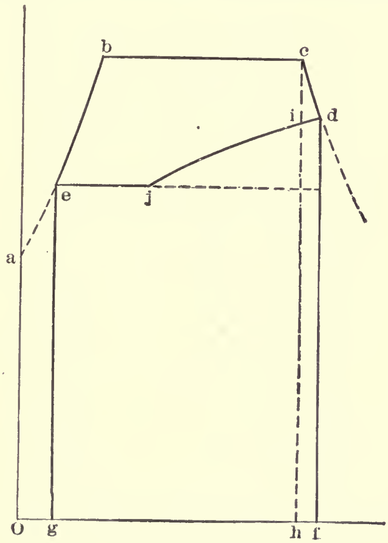


Fig. 205.

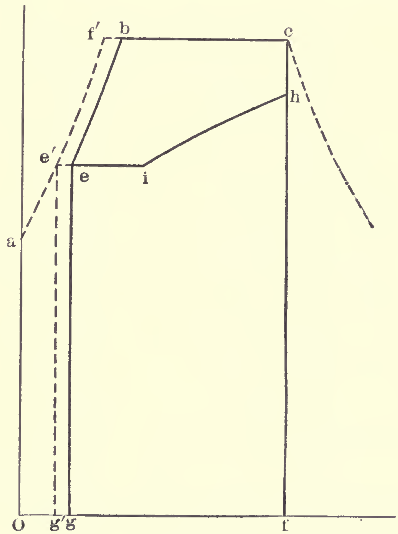


Fig. 206.

the efficiency of the heat supplied by the jacket is even less than when the expansion is complete.

In drawing the temperature-entropy diagram of the imperfect cycle with clearance, it is necessary to assume, as we did in the case of the Rankine cycle with complete compression, that the water in the boiler has the same temperature as the cushion steam at every instant during the compression period and that the sum of the weights of the cushion steam and the water in the boiler is 1 pound. By employing such assumptions the temperature-entropy diagram may be drawn, but this temperature-entropy diagram will not, of course, represent truly the actual condition of affairs in a real engine.

In Fig. 206 may be seen the temperature-entropy diagram of the imperfect cycle with complete compression. The line *e–b* is the compression line and is found in a manner similar to the line *e–b* in Fig. 202.

The work lost on account of clearance is, of course, the area *f′ b e e′*. However, on account of the cushion steam contained in the clearance spaces it is unnecessary to add the heat represented by the area *g′ e′ f′ b e g*. The ratio of the work lost to the heat saved is, however, greater than the ratio of the work done (area *e b c h i*) to the heat actually added (area *g e b c f*) and on this account the efficiency of the cycle is reduced by the use of clearance.

In Fig. 207 will be seen the temperature-entropy diagram of an imperfect cycle having clearance and no compression. The line *e–b* represents the rise in pressure due to the introduction of steam from the boiler, and is a constant volume line. It will be seen that the ratio of the

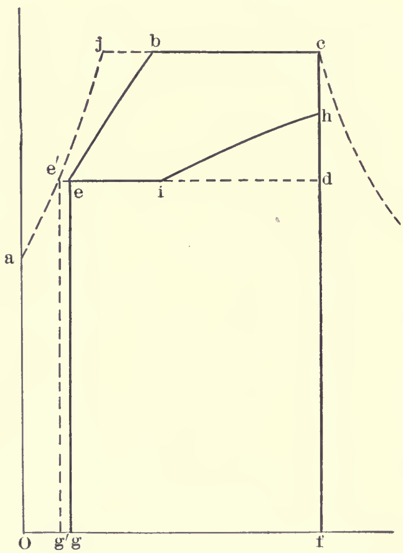


Fig. 207.

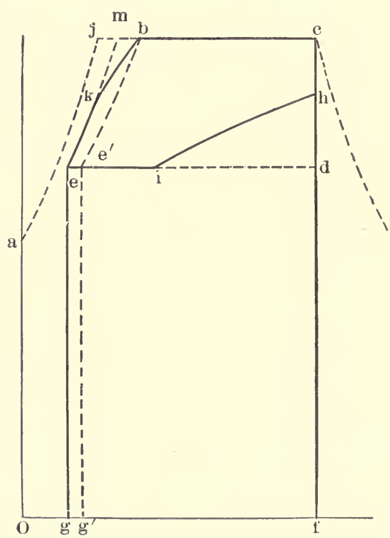


Fig. 208.

work lost on account of clearance, (area *j b e e′*) to the heat saved, (area *g′ e′ j b e g*) is much greater than the ratio of the work lost to the heat saved in Fig. 206. The use of compression therefore raises the efficiency of the cycle.

In Fig. 208 will be seen the temperature-entropy diagram of the imperfect cycle with partial compression. The compression line is line *e–k*, the admission line is line *k–b*, the steam line is line *b–c*, the expansion line is line *c–h*, the release line is line *h–i*, and the exhaust line is line *i–e*. The heat supplied is represented by the area *g e k b c f*, while the work done is represented by the area *e b c h i*. The area *i h d* represents the loss due to incomplete expansion. The area *k m b* represents the loss due to incomplete compression. This loss, however, cannot be diminished by

increasing the compression pressure, since, with the clearance volume shown by the diagram, if the compression pressure be raised until the compression is complete, the lost work due to clearance will be increased by the area $e'\ e\ k\ b$, while the heat saved will be represented by the area $g\ e\ k\ b\ e'\ g'$. If the ratio of the work lost to the heat saved is greater than the ratio of the work done to the heat supplied for the whole cycle, compression has been carried to too high a point.

344. The Temperature-Entropy Diagram for the Actual Steam Engine. In Fig. 209 will be found such a temperature-entropy diagram as would actually be obtained from a steam engine. The line a–b represents the period of admission. It will be seen that the temperature falls during this period on account of the fall in pressure due to wire drawing, and that throughout the admission period the steam line remains below the evaporation line g–h, which represents the temperature of the steam in the boiler. It is assumed that the steam is slightly wet as it enters the cylinder, so that the point h, which is the state point of the steam coming from the boiler, does not fall on the saturation line. The line b–c represents the expansion period. If the expansion were adiabatic, the line would be vertical. Actually the line is of the curved form shown. At the point where the admission ceases it will be noted that the line begins to run to the left, indicating that the steam is parting with heat more rapidly than it would as a result of adiabatic expansion. This is because cylinder condensation has not ceased at cut off, but continues until the temperature of the steam is below the average temperature of the surface of cylinder walls. At the point where the tangent to the expansion line is vertical, the rate of evaporation of steam from the clearance area

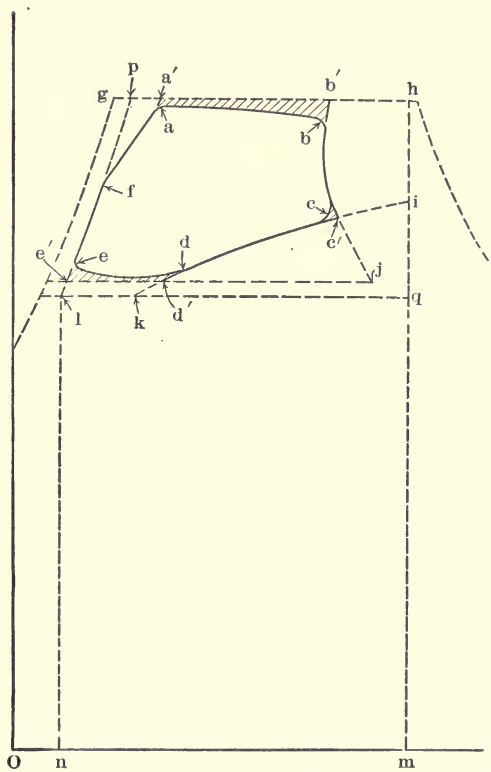


Fig. 209.

is equal to the rate of condensation upon that portion of the barrel which is just being uncovered by the moving piston, and from that point onward re-evaporation is more rapid than condensation. As a result, the line tends toward the right, indicating an addition of heat. At *c*, release occurs. The line *c–d* is not, however, a line of constant volume, since release does not occur at the end of the stroke, and the fall in pressure is gradual and not sudden on account of wire drawing through the exhaust ports. The line *d–e* represents the period of exhaust during which the back pressure remains practically constant. The line *e–f* represents the period of compression during which the pressure and temperature of the steam rises. The pressure and temperature of the cylinder feed in the boiler is assumed to rise simultaneously. As has been previously noted, the line *d–f* will not be a vertical line, in spite of the fact that the compression of the cushion steam is adiabatic, since that portion of the working fluid which is contained in the boiler is receiving additions of heat during the compression period. The line *f–a* is the admission line, and is of course, a constant volume line.

345. Graphical Analysis of the Losses in a Steam Engine. The actual temperature-entropy diagram for a steam engine may be superimposed upon the Rankine cycle temperature-entropy diagram in order to determine the amount and distribution of losses in the engine. In Fig. 209 $l\ p\ h\ q$ is the Rankine cycle diagram with complete compression for 1 pound of working fluid between the temperature limits of the boiler and the discharged condensing water. The quantity of heat supplied is $n\ l\ p\ h\ m$. Of this heat, the quantity $n\ l\ q\ m$ would be lost in the exhaust of the Rankine cycle, while the remainder would be transformed into work. Since the imperfect cycle is employed, that quantity of work represented by the area $k\ i\ q$ is lost on account of incomplete expansion and that quantity represented by $f\ p\ a'$ is lost on account of incomplete compression. It is unnecessary, however, to supply the quantity of heat represented by this small area, but since this entire quantity of heat would be transformed into work, a considerable loss is represented by its absence. The area $e'\ d'\ k\ l$ represents the loss due to imperfect condenser action. The area $b'\ h\ i\ c'$ represents the power loss with the given quantity of steam and the given terminal volume due to cylinder condensation. The area $d'\ c'\ j$ then represents the theoretical loss resulting from the fact that the actual expansion line is not continued to the back pressure line, following the same law of expansion as it does from b' to c'. Were the expansion continued to this point, however, as a result of the increased temperature range of the cylinder walls, the form of the temperature-entropy diagram would be changed, and the distribution of losses altered considerably. The shaded areas represent the loss which results from wire drawing and fluid friction and the ratio of the actual temperature-entropy diagram

to the diagram bounded by a' b' c' d' e' f is the card factor of the engine.

It must not be inferred that all heat losses shown by the temperature-entropy diagram are due to the direct transfer of heat to or from the steam. They may be due to the loss of steam from the working chamber on account of leakage. On the temperature-entropy diagram, as on the indicator card, leakage shows exactly the same effects as does cylinder condensation. We cannot therefore infer that the heat transfer shown occurs between working fluid and the cylinder wall.

346. Temperature-Entropy Diagram for Compound Engine. Usually the weight of the working fluid contained at cut-off in the high-pressure cylinder of a compound engine is different from that contained at cut-off in the low-pressure cylinder on account of the difference in the weight of the cushion steam. It is therefore impossible to draw a temperature-entropy diagram which represents the behavior of the steam in such an engine, although such a diagram may be drawn for each cylinder separately. The diagram of the low-pressure cylinder will fall below that of the high-pressure cylinder when they are superimposed on the same steam dome, and they may slightly overlap in case the indicator cards overlap. Since a temperature-entropy diagram is always drawn for 1 pound of working fluid, it follows that the weight of the cylinder feed shown by the two diagrams will be different, and it is necessary, therefore, to treat the losses occurring in each cylinder separately. The quantity of heat supplied to the second cylinder **per pound of cylinder feed** will be equal to the total quantity of heat rejected from the first cylinder **per pound of cylinder feed,** less the quantity of heat lost from the first cylinder by radiation, which is usually extremely small. Hence, in order to obtain the efficiency of a compound engine from such a combined temperature-entropy diagram, it is necessary to obtain the efficiency shown by each diagram separately, in which case the efficiency of the engine will be the efficiency shown by the high-pressure diagram plus the efficiency shown by the low-pressure diagram, multiplied by one minus the efficiency shown by the high-pressure diagram. The temperature-entropy diagram for a compound engine may be employed in order to analyze the heat transfers occurring within each of its cylinders, and will show the causes and relative amounts of the losses in each of the cylinders.

347. Transferring an Indicator Diagram to the Temperature-Entropy Plane. In order to employ the temperature-entropy diagram to illustrate graphically the distribution of losses in the steam engine, it is necessary to have an indicator card representative of the average conditions in the two ends of the cylinder of the engine for the entire test, and to transfer it to the temperature-entropy plane. In Fig. 210 such a card is shown in the quadrant $P\ O\ V$. In the quadrant $P\ O\ T$ is the pressure-

temperature curve of saturated steam. In the quadrant $T\ O\ N$, which is the temperature-entropy plane, the desired temperature-entropy diagram is to be drawn. In the quadrant $V\ O\ N$ is drawn the line $H\ H$, which gives the relation between the volume and the entropy of 1 pound of water. Since the distance of line $H\ H$ from line $O\ N$ is extremely small as compared with the other distances to be measured in this quadrant, no serious error will be introduced if this line is omitted and the constructions are based upon line $O\ N$ instead of line $H\ H$. The saturation curve for the quantity of steam which the test shows to be contained in the cylinder per revolution at cut-off, is next drawn in the quadrant $P\ O\ V$, and is the line $R\ Q$. In the quadrant $T\ O\ N$ are drawn the lines $O\ W$ and $S\ N$, the first being the water line and the second the saturation line. In order to transfer any point on the indicator card, as, for instance, point B, to the temperature-entropy plane, the following construction is employed. Draw line $A\ C$ through B, then draw lines $A\ D$ and $C\ E$. Next draw lines $E\ D$ and $G\ H$, then $D\ H$, then $B\ F$, and finally $F\ I$. The intersection of $C\ E$ and $S\ N$ determines the entropy of 1 pound of dry and saturated steam at the temperature represented by point B. The entropy of evaporation of the wet steam at point B is proportional to the increase in its volume which results from its evaporation (i.e., to its quality). By the construction, we have divided the evaporation line $G\ E$ into two segments, one of which, $G\ I$ bears the same ratio to $G\ E$, as the volume of the steam at B does to the specific volume of dry and saturated steam. The point I is therefore the state point on the temperature-entropy diagram of the point B on the pressure-volume diagram. Other points on the temperature-entropy diagram may be found in the same way, and the diagram drawn.

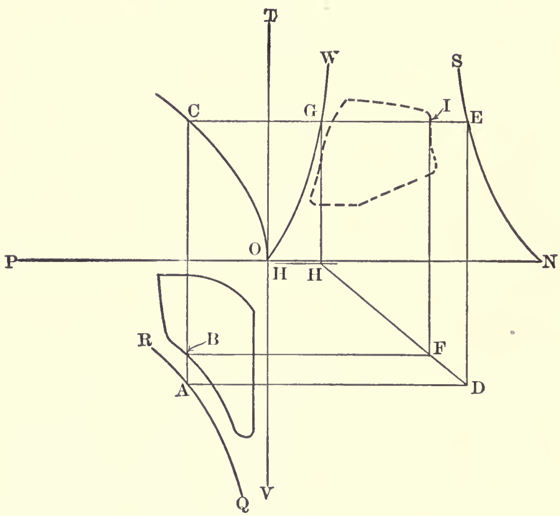


FIG. 210.

348. The Temperature-Entropy Diagram for the Steam Turbine. Since a single-stage steam turbine operates upon the Rankine cycle, the theoretical temperature-entropy diagram will be that of the Rankine cycle

for steam of the quality supplied to the turbine. Turbines are usually supplied with superheated steam so that the temperature-entropy diagram will usually have the form shown in Fig. 201. However, after the steam has passed through the nozzle of the turbine, it loses a portion of its kinetic energy, which is retransformed into heat by eddying and fluid friction, so that although the kinetic energy of the steam flowing from a nozzle is in theory equal to the area *e b c h d*, the work which actually is transferred to the rotating member is much less.

Actually, the expansion of the steam in the turbine nozzle is not quite adiabatic, since on account of friction, its entropy is continually increased by the retransformation of a portion of its kinetic energy into

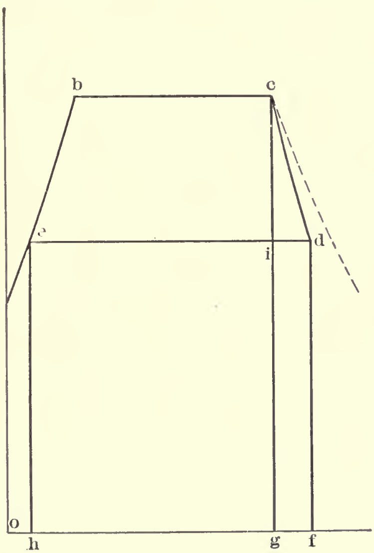


Fig. 211.

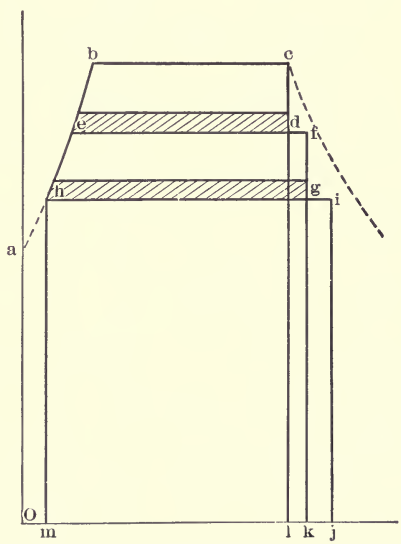


Fig. 212.

heat. The diagram which represents the condition of affairs in such a nozzle is shown in Fig. 211, in which line *c d* represents the relation between the temperature and the entropy of the expanding steam. The kinetic energy of the jet proceeding from the nozzle is not, however, equal to the area *b c d e*, since a portion of the work generated was transformed into the heat represented by the area *c d g f*. The actual kinetic energy of the issuing jet is equal to the area *b c d e* minus the area *i d f g*. The amount of heat represented by the area *i d f g* is exceedingly small, however, when the nozzle is properly designed, and its effect in modifying the temperature-entropy diagram is almost imperceptible. The heat supplied in the entering steam is, of course, represented by the area *h e b c g*.

In Fig. 212 may be seen the temperature-entropy diagram of a two-stage impulse turbine. A part of the kinetic energy of the jet issuing form the nozzles in the first stage is retransformed into the quantity of heat represented by the area *f k l d*. The equivalent area is shown subtracted from the work of the first stage and is represented by the shaded area within *e b c d*. In like manner, in the second stage, a portion of the work *e f g h* is retransformed into the quantity of heat represented by the area *g i j k* and is shown by the shaded area taken out of *e f g h*. The heat supplied in the entering steam is represented by the area *m h b c l*, and the useful work is measured by the unshaded area included within *h b c d f g*.

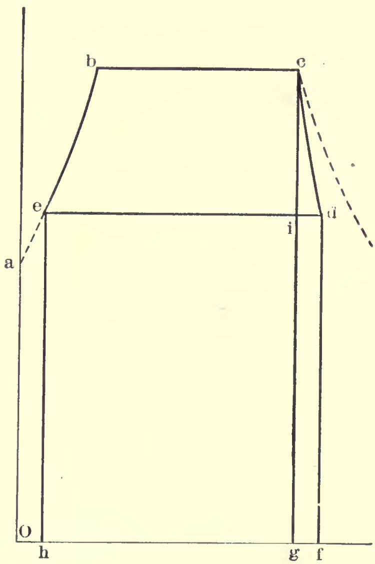


FIG. 213.

In the case of a turbine containing a very large number of stages, whether it be an impulse turbine or an impulse reaction turbine, the temperature-entropy diagram will have approximately the form shown in Fig. 213. The quantity of heat added is that represented by the area *h e b c g*. A portion of the kinetic energy of the steam flowing in the vanes is retransformed into the quantity of heat represented by the area *c d f g* and the relation between the temperature and entropy of the steam as it traverses the turbine is represented by the line *c d*. The total amount of heat transformed into work is represented by the area *e b c d* minus the area *i d f g*.

349. The Total Heat-Entropy or Mollier Diagram. It is very convenient in designing thermodynamic apparatus, particularly in the case of steam turbines, to make use of the Mollier diagram, which gives graphically the relation between the total heat and total entropy of steam. Such a diagram accompanies both Marks and Davis' and Peabody's steam tables, and is shown in skeleton in Fig. 214. On this diagram are drawn lines of constant quality (such as *a-b*) or constant superheat (such as *c–d*) and lines of constant pressure (such as *e–f* or *g–h*) which are, of course, lines of constant temperature in that portion of the diagram which represents the properties of wet steam. On this diagram, vertical lines are lines of constant entropy and are therefore adiabatic lines. Horizontal lines are lines of constant total heat. Any point on the diagram is determined

by the intersection of four lines, namely a constant total heat line (horzontal) a constant entropy line (vertical), a constant pressure line (diagonal) and a constant quality or superheat line (sloping). If, for instance, the pressure and quality of steam are known, its total heat and entropy may be determined. For instance, point *m* is the state point on this diagram of steam of 21 pounds pressure and 80 per cent quality. The ordinate to this point gives the total heat of the steam (967 B. T. U.), and the abscissa its entropy (1.45). If any two of the four properties are known, the other two may be determined from the diagram.

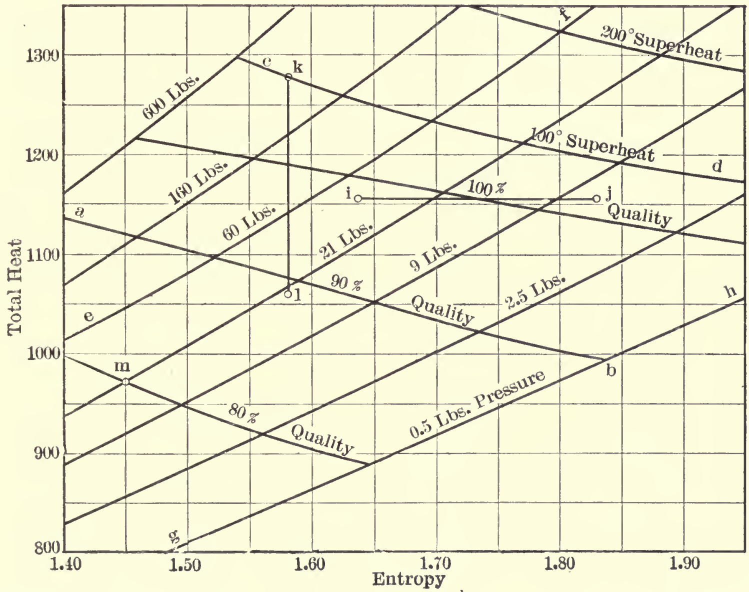


FIG. 214.

In order to use such a diagram in the solution of problems, it is convenient to paste the diagram on a well-seasoned drawing board, and to cover it with a sheet of transparent celluloid, mounting it carefully, so that its coordinates are vertical and horizontal.

By graphical construction upon such a diagram, many problems may be solved very quickly. For instance, let it be assumed that in a throttling calorimeter the temperature and pressure of the steam are those shown by point *J* (this would be possible in case the calorimeter were connected to a condenser in order to increase the permissible wetness of the steam to be tested). By drawing through *J* a horizontal (i.e., a constant total heat) line, the intersection of this line with the pressure line of the steam in the pipe from which the sample was taken (as at *i*), will give the state point of that steam, and its quality will be known.

In like manner if point k be assumed to be the state point of the steam entering a nozzle, as determined from its pressure and superheat, a vertical (i.e., an adiabatic) line drawn to intersect the pressure line which represents the pressure of the steam as it issues from the nozzle, will determine the properties of steam. If point l is at the intersection of the adiabatic and the terminal pressure line, it is the state point of the issuing steam. The length k–l will then represent the heat transformed into kinetic energy, and by transferring the distance to the velocity scale at the left of Marks and Davis' diagram, the velocity may be read off directly, the final velocity being determined by the distance from the initial velocity of the steam entering the nozzle. Other constructions will readily suggest themselves to the reader for solving various problems in connection with the properties of steam or its behavior in engines and turbines.

350. Temperature-Entropy Diagrams for Hot-air Engines. The temperature-entropy diagram may be used to illustrate the action of the working fluid in a hot air engine. In Fig. 215 will be found the temperature-entropy diagram of a Joule cycle engine. The line $d\ a$ represents the adiabatic compression of the working fluid, the line a–b the increase in entropy and temperature which occurs in the heater, the line b–c represents the adiabatic expansion of the working fluid, and the line c–d, the decrease in temperature and entropy which occurs in the cooling chamber. Lines a–b and d–c will, of course, be logarithmic curves, since the specific heat of the gas is constant. The quantity of heat supplied is represented by the area $f\ a\ b\ e$, the quantity of work done by the area $a\ b\ c\ d$ and the heat rejected by the area $d\ c\ e\ f$. It will be seen that the efficiency of the cycle will be increased by increasing the temperature range during the adiabatic expansion or compression and that the lower the initial temperature at the beginning of adiabatic expansion, the greater will be the efficiency of the cycle.

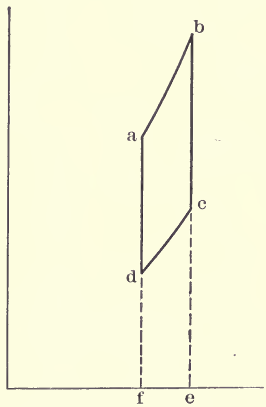


Fig. 215.

No temperature-entropy diagram can be drawn for a Stirling cycle engine, since the conditions of operation of the engine are contrary to the fundamental assumption made in drawing a temperature-entropy diagram. In the Stirling engine, a portion of the working fluid is at the temperature of the heater while the remainder of it is at the temperature of the cooler. The temperature-entropy diagram assumes that the entire quantity of the working fluid is at the same temperature, consequently no temperature-entropy diagram can be drawn for this engine.

351. The Temperature-Entropy Diagram of the Otto Cycle. The temperature-entropy diagram of the Otto cycle may be seen in Fig. 216. It is bounded by two adiabatics, *a–b* and *c–d*, and two constant-volume lines *b–c* and *a–d*. These lines are, of course, logarithmic curves. It is assumed that the specific heat of the working fluid is constant. The heat supplied is, of course, the area *b–c–e–f*, and the work done is the area *b c d a*, and its heat rejected is area *f a d e*.

352. Temperature-Entropy Diagram of other Gas-engine Cycles. The temperature-entropy diagram of the Sargent cycle may be found in Fig. 217. *a–b* and *c–d* are adiabatics, *b–c* is a constant-volume line, *d–e* is a

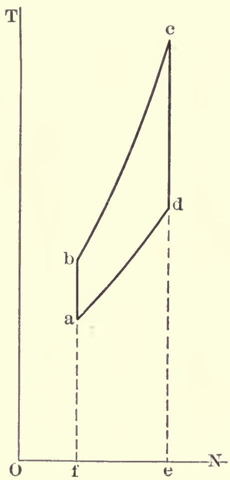


FIG. 216.

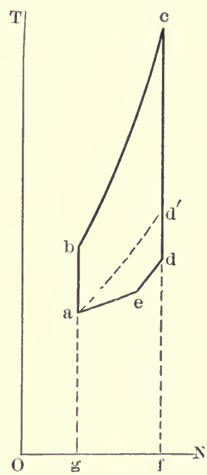


FIG. 217.

constant-volume line and *e–a* is a constant-pressure line. The heat supplied is *b c f g* and the work done *b c d e a*. The extra power gained over the Otto cycle is represented by the area *d′ a e d*, line *a–d′* being a constant volume line.

The temperature-entropy diagram of the Diesel cycle with isothermal expansion is seen in Fig. 218. Lines *a–b* and *c–d* are adiabatics, *b–c* is an isothermal, and *d–a* a constant-volume line. In Fig. 219 is seen the temperature-entropy diagram of the Diesel cycle with isobaric expansion. *a–d* and *c–b* are adiabatics, *d–c* is the isobaric line and *b–a* the constant-volume line. It will be seen that by increasing the temperature range by isobaric instead of isothermal expansion, the theroretical efficiency of the cycle is increased.

353. The Actual Temperature-Entropy Diagram of an Otto Engine. The form of temperature-entropy diagram which will actually be given by an Otto cycle engine may be seen in Fig. 220. The line *a–c* is

the actual compression line. The line c–x, which is, of course, approximately a logarithmic curve, represents the rise in temperature at explosion. The line x–t is the expansion line and the line t–a represents the fall in pressure at the end of expansion. This diagram is shown superimposed upon the diagram $a\ c'\ x'\ t'$, which is the theoretical temperature-entropy diagram for an Otto cycle engine when the quantity of heat added at explosion is the entire quantity of heat contained in the working fluid. The diagram $a\ c'\ x''\ t''$ is the theoretical temperature-entropy diagram of the Otto cycle for the same temperature limits as occur in the actual engine.

It will be noted that the actual compression line a–c at first runs to the right and then turns off to the left. The reason for this is that during

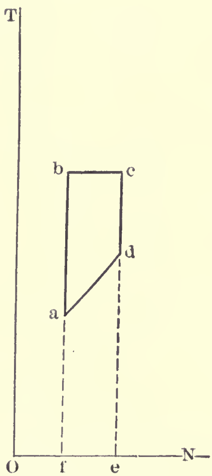


FIG. 218.

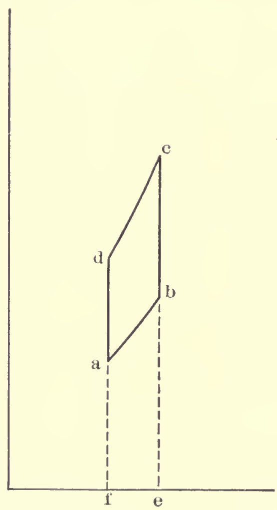


FIG. 219.

the early part of the compression stroke the temperature of the walls is higher than that of the working fluid, the working fluid receives heat from the walls, and its entropy is increasing. The working fluid soon becomes hotter than the walls, however, and begins to lose heat to them, so that its entropy decreases during the latter portion of the compression stroke. During the explosion the working fluid is receiving heat on account of the combustion of the charge, and is losing heat to the cylinder walls. The line c–x represents the net effect of the addition and abstraction of the heat during this period. During the expansion period the working fluid is receiving heat on account of the delayed combustion of the charge, and it is also losing heat to the cylinder walls. On the whole the quantity of heat lost to the walls is greater than that received from

the combustion of the charge, so that the expansion line tends to the left as it descends. The line t–a is of course identical with the theoretical line.

The exact form of temperature-entropy diagram given by an Otto cycle engine will vary with the size of the engine, the character of the charge, and the relative amounts of the several losses. The temperature-entropy diagram is not as useful in analyzing the losses occurring in a gas engine as it is in analyzing those in a steam engine, and care must be taken when employing the diagram to see that the quantities of heat represented on the diagram are properly interpreted.

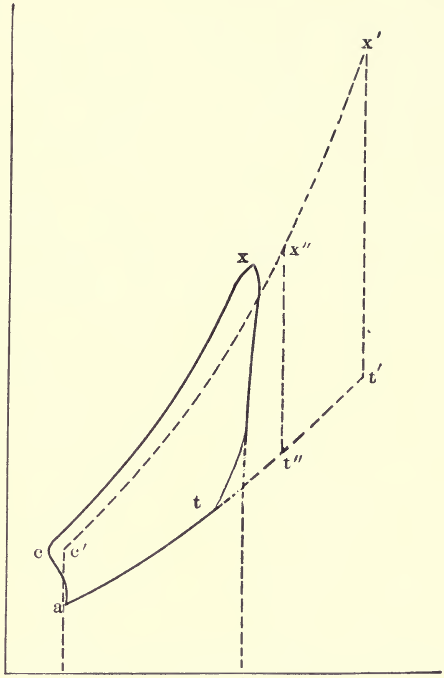


Fig. 220.

354. The Air Compressor. No temperature-entropy diagram can be drawn for an air-compressor cycle, since the quantity of working fluid contained in the cylinder is not constant. It might be thought that the diagram could be constructed by drawing two adiabatics and two constant-pressure lines for air, but it must be borne in mind that the air is expelled from the cylinder at constant temperature as well as at constant pressure. The constant-pressure line of the temperature-entropy diagram assumes that the weight of the working fluid is constant, and that its volume is decreased by changing its temperature. It will be seen then that a temperature-entropy line diagram cannot be drawn, since the working fluid does not perform a true cycle.

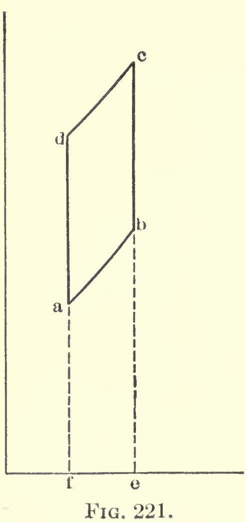


Fig. 221.

355. The Temperature-Entropy Diagram of Refrigerating Machines. The theoretical temperature-entropy diagram for a refrigeration cycle is similar in its general appearance to that of a heat-engine cycle. The temperature-entropy diagram of the reversed Joule cycle may be seen in Fig. 221. Line a–b represents the absorption of the quantity

of heat measured by the area *f a b e* in the vaporizer, which raises the temperature of the working fluid from point *a* to point *b* and increases its entropy. Adiabatic compression then increases its temperature to point *c* without changing its entropy. The line *c–d* represents the process of cooling the working fluid in the condenser at constant pressure, during which time it parts with the quantity of heat represented by the area *f e c d*. The area *f e c d* represents the heat taken from the vaporizer, *f e b a*, plus the work done upon the working fluid, *a b c d*.

The temperature-entropy diagram of the ammonia-compression cycle may be seen in Fig. 222. At point *a* compression of dry and saturated ammonia vapor begins. The ammonia vapor in practice may be slightly wet or slightly superheated, but it is usually nearly dry and saturated. This vapor is compressed adiabatically along the line *a–b* and becomes superheated during the process. The superheated vapor is introduced into the condenser where it is condensed at constant pressure (after being cooled to the saturation temperature) along the constant-pressure line *b–c–d*. In passing through the expansion valve, the total heat of the ammonia vapor is unchanged, and the fall in temperature is therefore represented by the constant total heat line *d–e*. The entropy of evaporation of that portion of the liquid which is evaporated by the heat of the liquid, is represented by the segment *O–e*. The remainder of the latent heat of evaporation is taken from the vaporizer at constant temperature, the process being represented by the line *e–a*. The heat abstracted from the vaporizer is, of course, represented by the area *g e a f*, while the heat rejected to the condenser is equal to the area *g e d c b f*.

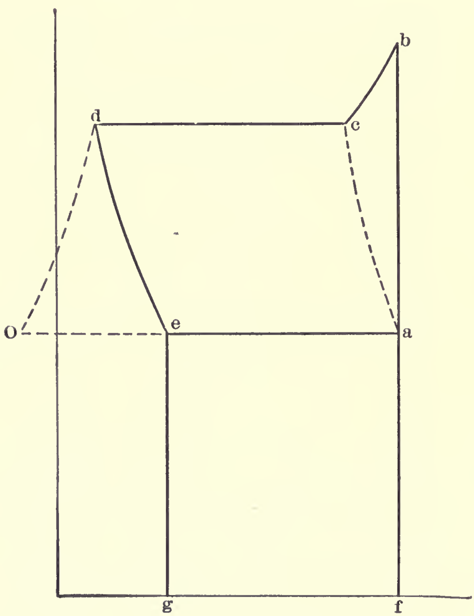


FIG. 222.

The actual temperature-entropy diagram which would be obtained from an ammonia compression plant may be seen in Fig. 223. The vapor that comes to the compressor is usually slightly wet. Since the cylinder of the machine is warmer than the vapor which it compresses, this vapor is almost immediately dried and then continues to gain heat during the

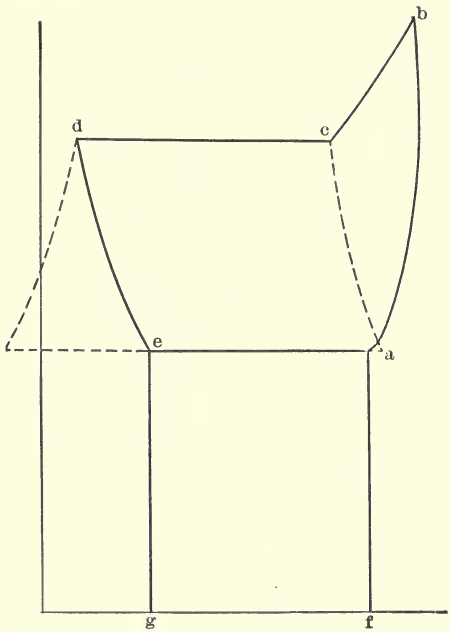


FIG. 223.

most of the compressed period. The remainder of the diagram is identical with the theoretical diagram, but the compression line a–b is approximately of the form shown. Since the ammonia vapor is not entirely dry as it enters the cylinder, the amount of heat abstracted from the vaporizer per pound of working fluid, represented by the area $g\ e\ a\ f$, is less than would be transferred if the ammonia were dry. At the same time, the amount of work required to compress the ammonia and deliver it into the condenser is considerably greater than it was in the theoretical cycle, since the vapor gains heat during the early part of the compression period from the walls of the compressor cylinder. The efficiency of the system is, of course, equal to the area $g\ e\ a\ f$ divided by the area $e\ d\ c\ b\ a$.

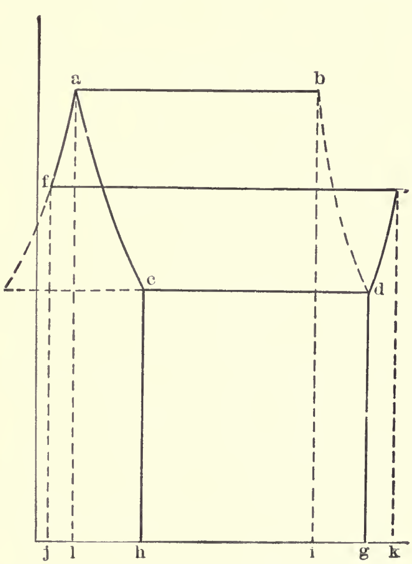


FIG. 224.

The temperature-entropy diagram for an absorption system may be seen in Fig. 224. In this diagram, line a–b represents the vaporization of the ammonia at constant pressure in the generator, while line b–a (the same line reversed) represents its condensation in the condenser. The ammonia liquid now escapes through the expansion valve, a process represented by the constant total heat line a–c. Within the vaporizer, the remainder of the liquid evaporates, a process represented by the evaporation line c–d. This ammonia is then superheated by absorbing heat at constant pressure along the line d–e. It then condenses by absorption in water and

the heat is removed from the absorber at constant temperature, that process being represented by the line *e–f*. Its temperature is now raised by the application of heat until the liquid is sufficiently hot, so that the ammonia may be vaporized. The heat absorbed in the vaporizer is represented by the area *c d g h*. The heat supplied to the generator is represented by the area *l a b i*. The heat taken from the absorber is represented by the area *j f e k*. The heat represented by the area *j f a l* is returned by the regenerator coil, from the spent liquid entering the absorber, to the liquid entering the generator. The quantity of heat absorbed by the condenser is represented by the area *l a b i*.

In drawing this diagram it has been assumed that the heat added or abstracted was the latent heat of ammonia vapor. Since, however, ammonia has an affinity for water, additional quantities of heat are transferred, due to the extra heat required to evaporate ammonia from an aqueous solution. No account, however, has been taken of these quantities in this diagram.

CHAPTER XXVI

THE KINETIC THEORY OF HEAT

356. The Kinetic Theory of Gases. All of the phenomena noted in the first seven chapters of this book, and collectively termed the properties of gases and vapors and their mixtures, may be entirely explained by the kinetic theory of heat. This theory assumes that any mass of gas or dry vapor is composed of a great number of very small particles. These particles are perfectly elastic, and in the case of a perfect gas, exert no force upon anything with which they are not in actual contact. In the case of imperfect gases, these particles tend to attract or repel one another. The mass of each particle, although minute, is definite and unchangeable. The volume of each particle is infinitesimal in the case of a perfect gas. In the case of an imperfect gas, the volume is finite, although extremely minute. In case the gas is of homogeneous composition, each of the particles is exactly like every other. These particles are in chemistry termed molecules, and hereafter in this chapter will be designated by that term.

Assume that a number of such molecules are confined within the walls of a vessel; and that these walls are motionless and absolutely rigid. Each particle, once it is set in motion, will move with uniform velocity in a right line until it impinges upon one of the walls. Since it is perfectly elastic, and the wall is perfectly rigid, it will rebound from the wall with undiminished velocity and unchanged energy. It will proceed in its path until it encounters a second wall, from which it will rebound in the same manner, and will continue thus to travel in straight lines from wall to wall with unchanged velocity. Each of the particles contained in the vessel will behave in the same manner, and since they are assumed to be infinitesimal in size, they will encounter each other only at infinite intervals. As the result of the impact upon the walls of the vessel, these walls will experience a pressure whose amount we will now determine.

357. Pressure Exerted on Account of the Motion of the Molecules. Assume that the vessel is a cube bounded by six planes, each 1 foot square. It will then contain 1 cubic foot of gas. Assume that this volume contains a molecule whose mass is m, and whose velocity, normal to a given face, is V_1 feet per second. Since the distance from the given face to the opposite face and back is 2 feet, the molecule will strike the given face $\frac{V_1}{2}$ times per second. Each time it strikes the face its momentum is changed by the amount $2mV_1$. Each second, the given face will change the momentum of the molecule by the amount $m\ V_1^2$. The rate of change of momentum is numerically equal to the force producing the change. Hence the mean value of the force exerted by the face in resisting the impacts of this molecule is $m\ V_1^2$.

If the space contains, in addition, a second molecule having the same mass, and whose velocity normal to the given plane is V_2, then the force exerted by the impacts of the two molecules will be $m(V_1^2 + V_2^2)$, and so on for any number of molecules. Finally, if the space contains n molecules, the force exerted by their impacts upon the face will be

$$P = m(V_1^2 + V_2^2 + \ldots + V_n^2). \quad \quad (1)$$

Designating the product of m and n by M, and

$$\frac{(V_1^2+V_2^2+\ldots+V_n^2)}{n} \text{ by } \overline{V}_n^2,$$

we will have for the pressure in pounds per square foot exerted by the gas

$$P = M\overline{V}_a^2. \qquad (2)$$

In the case of an actual gas under an appreciable pressure the molecules are very great in number and are traveling in all directions. The value of $\overline{V}_n^2$ is therefore the same for each of the six faces of the cube. The square of the velocity of each particle is the sum of the squares of the velocity components normal to each of three adjacent faces. Consequently, the mean of the squares of the velocities of all the molecules is three times the mean of the squares of any component of the velocities. We may therefore write

$$\overline{V}^2 = 3\overline{V}_n^2. \qquad (3)$$

Replacing $\overline{V}_n^2$ by $\frac{\overline{V}^2}{3}$,

and M by $\frac{W}{g_0}$ in equation (2), we will have

$$P = \frac{W\overline{V}^2}{3g_0} \qquad (4)$$

in which W is the weight of the gas in pounds per cubic foot, $\overline{V}^2$ is the mean of the squares of the velocities of all the molecules, and g_0 is the standard acceleration of gravity. Since the weight of the gas per cubic foot is equal to the weight per molecule (w) multiplied by the number of molecules per cubic foot (n) we may write the above equation in the form

$$P = \left(\frac{w\overline{V}^2}{2g_0}\right) \times \left(\frac{2n}{3}\right). \qquad (5)$$

In the above expression, $\frac{w\overline{V}^2}{2g_0}$ is of course the mean kinetic energy per molecule of the gas. Hence the pressure of the gas is proportional to the mean kinetic energy per molecule, and to the number of molecules per unit of volume. Substituting $\frac{1}{3}\overline{V}^2$ for $\overline{V}_n^2$ in equation (2), we will have

$$P = \tfrac{1}{3} M\overline{V}^2, \qquad (6)$$

when the mass of gas M occupies one cubic foot. If it occupies V cubic feet, we will have

$$PV = \tfrac{1}{3} M\overline{V}^2, \qquad (7)$$

in which P is the pressure of the gas in pounds per square foot, V is the volume of the gas in cubic feet, M is the mass of the gas in kinetic mass units, and is equal to $\frac{W}{g_0}$, where W is its weight in pounds, and $\overline{V}^2$ is the mean of the squares of the molecular velocities.

358. Temperature of a Gas Proportional to the Mean Kinetic Energy of the Molecules. If we increase the mean kinetic energy per molecule, we must do so by transferring energy to the gas. If we assume that this energy is transferred to the gas in the form of heat, we will have the mean kinetic energy per molecule increased by

the addition of heat. We know that the addition of heat to a gas increases its temperature and pressure, and if the gas be perfect, the increase in temperature and pressure is proportional to the quantity of heat added. Hence we may conclude that the absolute temperature as well as the pressure of any gas given is proportional to the mean kinetic energy per molecule.

A further inspection of equation (5) in the preceding article will show that if we change the mass of a molecule without changing the number of the molecules, that the pressure of the gas will be the same when $\overline{V}^2$ is so changed as to make the mean kinetic energy per molecule the same. Certain chemical phenomena point to the conclusion that the number of molecules in a given volume of gas confined at a given pressure and temperature is always the same whatever be the nature of the gas. Hence, we arrive at the conclusion that the absolute temperature of any gas is proportional to the mean kinetic energy per molecule.

When a cubic foot of gas is heated at constant volume, the heat will be entirely expended in increasing the velocity of the molecules. Consequently, the heat imparted to the gas, when measured in dynamic units, will be

$$\Delta H_v = \frac{M\overline{V_2}^2}{2} - \frac{M\overline{V_1}^2}{2}. \qquad (8)$$

in which $\overline{V_2}^2$ is the mean of the square of the final molecular velocities, and $\overline{V_1}^2$ is the mean of the squares of the initial velocities. This equation may be written

$$\Delta H_v = \frac{M}{2}(V_2^2 - V_1^2). \qquad (9)$$

359. Loss of Energy during Expansion. When the gas is heated at constant pressure, a part of the energy is expended in doing external work. If we assume that all of the energy is first transferred to the molecules, then some of this energy will be surrendered by the molecules during their impact upon the moving wall of the expanding vessel. The increase in volume of the gas is proportional to the absolute temperature and is consequently proportional to the change in the mean square of the velocity of the moelcules. The work done by the gas is found by multiplying the initial pressure by the change in volume. Since the initial volume is 1 cubic foot, and the volumes are proportional to the mean of the squares of the velocities of the molecules, we will have for the change in volume

$$\Delta V = \frac{\overline{V_2}^2 - \overline{V_1}^2}{\overline{V_1}^2}. \qquad (10)$$

For the pressure of the gas we will have the value

$$P = \frac{m\overline{V_1}^2}{2} \times \frac{2n}{3}. \qquad (11)$$

Multiplying (10) and (11) together we will have for the work of expansion

$$P\Delta V = (\overline{V_2}^2 - \overline{V_1}^2)\frac{M}{3}. \qquad (12)$$

The increase in the kinetic energy of the molecules is given by equation (9), Art. 358. Adding this to equation (12), we will have for the energy required to heat a cubic foot of the gas at constant pressure

$$(\overline{V_2}^2 - \overline{V_1}^2)\frac{5M}{6} = \Delta H_p. \qquad (13)$$

The quantity of energy given by equation (9) Art. 358, is proportional to the specific heat of the gas at constant volume. That given by equation (13), is proportional to the specific heat of the gas at constant pressure. Dividing the latter by the former we will have for the ratio of the specific heats

$$\frac{5\,M}{6} \div \frac{M}{2} = 1\tfrac{2}{3} = \gamma. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

We may therefore conclude from equation (14) that the value of γ for a gas whose molecules have translational energy only, is always $1\tfrac{2}{3}$.

360. Intra-molecular Energy. The molecules of a gas will have translational energy only, when each of the molecules consists of one particle whose dimensions are infinitesimal. Such a gas is said to be monatomic. If the molecule is composed of two or more particles (in chemistry termed atoms) with a finite distance between them, it will be apparent that they must have a motion relative to one another, which will absorb a portion of the heat energy imparted to the molecule. Such a gas is said to be polyatomic. The energy which the molecule contains in virtue of the relative motion of its atoms is termed **intramolecular energy.** The energy which it contains in virtue of its mass and the velocity of its center of gravity, is termed **translational energy.** The sum of the intramolecular and translational energy of the molecule is termed the **intrinsic energy.** Clausius has shown that the ratio of the mean intramolecular energy to the mean translational energy is a constant for any gas. This ratio is therefore called Clacsius' ratio, and is designated by the letter ρ.

The energy which must be imparted to the gas for the purpose of increasing its temperature will be $1+\rho$ times that which would be required by the same volume of a monatomic gas. We will have then for the quantity of energy imparted to the gas in heating it, at constant volume,

$$JH_v = (1+\rho)\frac{M}{2}\,(V_2{}^2 - V_1{}^2). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

Adding equation (15) to equation (12) of the preceding article, we will have the energy imparted to the gas to heat it at constant pressure, which is

$$JH_p = \frac{5+3\,\rho}{3}\left(\frac{M}{2}\right)(V_2{}^2 - V_1{}^2). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

Dividing 16 by 15 we will have

$$\gamma = \frac{5+3\,\rho}{3+3\,\rho}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

It may be noted that when ρ_0 becomes zero, the value γ becomes $\tfrac{5}{3}$, as it should. Solving (17) for ρ, we will have for the value of Clausius' ratio for a gas for which the constant γ is known,

$$\rho = \frac{5-3\,\gamma}{3\,\gamma-3}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

361. Adiabatic Expansion of Gases. When a gas is confined within an expanding vessel, for instance, a cylinder having a moving piston, the mean intrinsic energy per molecule will diminish, since the molecules which strike the piston will rebound from its face with diminished **absolute** velocity, their velocity **relative** to the face of the piston being unchanged in amount and reversed in direction by the impact. The energy thus surrendered by the molecules reduces their mean kinetic energy and so reduces the temperature of the gas. In case the volume of the cylinder is diminished, the mole-

cules will rebound from the moving piston with increased velocity, the velocity relative to the face again being the same after impact as before. The result of such compression is, of course, to increase the mean kinetic energy per molecule and the temperature of the gas. The phenomena of adiabatic expansion and compression are thus fully explainable by the kinetic theory.

362. Condition of Equilibrium between Molecules of Different Masses. If a gas be conceived to consist of molecules of two different kinds (i.e., if it is a mixture of different gases) the molecules will pass among each other and occasionally molecules of one kind will, by collision or otherwise, exert force upon molecules of the other kind. Each time that force is exerted between two molecules of different kinds, they will exchange a portion of their kinetic energy. In general, the molecule having the greater kinetic energy will give up a portion of its energy to the molecule having less energy, so that as a result of the continued interchange of energy, we will find that the mean kinetic energy per molecule of the two different kinds of molecules will become equal, or in other words, the two constituents of the mixture will come to the same temperature.

In the case of such mixtures, the mean velocities of the different classes of molecules will be different. In order that the temperature of the two constituents shall be the same, the mean value of the kinetic energy per molecule must be the same for the several constituents. In order to make this true, it is necessary that the mean square of the velocities of each kind of molecules shall be inversely proportional to the mass of a molecule of that kind, so that the lighter molecules in a mixture will have high velocities while the heavier ones will have low velocities. We have already seen in equation (7) Art. 357, that $P\ V = \frac{1}{3}\ M\ \overline{V}^2$; this equation may be written

$$P\ V = \frac{W}{3\,g}\overline{V}^2. \qquad (19)$$

Substituting the value of $P\ V$ from the characteristic equation of gases, we will have, $\frac{W\ \overline{V}^2}{3\,g} = W\ R\ T$, whence we deduce that

$$\sqrt{\overline{V}^2} = \sqrt{3\,g\,R\,T}, \qquad (20)$$

which is an equation giving the value of the square root of the mean square of the velocities of the molecules of the gas. Substituting the proper values in this equation we will find that the velocity of air molecules at 70° F. will average about 1650 feet per second, while that of hydrogen molecules at the same temperature will average about 6250 feet per second. It will be seen that the average velocities of these molecules is quite high and a small proportion of them will have velocities far exceeding these average velocities. In case the velocity of a molecule exceeds about 35,000 feet per second, it will be carried beyond the sphere of the earth's attraction. It will thus be seen that the tendency will be for the lighter gases to gradually escape from the earth's atmosphere. Were the value of the attraction of gravitation sufficiently reduced, as it is in the case of the smaller heavenly bodies, the escape of gases would be made much easier, so that those bodies which are small as compared with the earth will have no atmosphere. The moon is such a body.

363. The Flow of Gas from a Vessel through an Orifice. Assume two vessels to be separated from one another by a partition. Assume that the right-hand vessel is filled with gas and that the left-hand vessel is empty. If now a small opening be made in the partition, a part of the gas will pass from the right-hand vessel into the left-hand vessel, until the pressures are equalized. The molecules contained in the right-hand vessel do not all have the same velocity, some of them having higher velocities than

others. Those having the higher velocities will rebound from the walls more often than do the others, and hence will have a better chance of passing through the opening. Consequently, after the gas has flowed from the right to the left, those molecules contained in the left-hand vessel will on the average have higher velocities than those contained in the right-hand vessel. The pressures in the two vessels will be equal, but the temperature of the gas contained in the left-hand side will be higher, and its mass will be less than that of the gas contained in the right-hand side, and the total energy (or heat) of the whole mass will be unchanged. In the course of time the entire mass of gas will gradually come to thermal equilibrium by the passage of molecules back and forth through the orifice, and finally each vessel will be filled with an equal number of molecules having the same mean energy per molecule.

If a current of gas be caused to flow through a porous plug, the temperature of the gas will be unchanged, since every portion of the gas as it is forced through the plug remains together and the molecules having different velocities are not permitted to pass into separate regions. If there are no forces exerted by the molecules upon one another, the mean kinetic energy per molecule and therefore the temperature of the gas will remain unchanged in passing through such a porous plug.

364. Osmosis. If a vessel to be assumed to contain two kinds of gas mixed together in equal proportions, the velocities of the molecules of the lighter gas will on the average, be much higher than those of the molecules of the denser gas. Molecules having high velocities will rebound more often from the walls than do the molecules having low velocities, and consequently will have greater opportunity for passing through an opening. If an opening be made in the wall of the vessel, the molecules of the lighter gas will pass through the openings more often than do those of the denser gas. If the opening connects the vessel with another similar but empty vessel in the manner described in the previous article, after the pressures are equalized the gas contained in the left-hand vessel will consist of a larger proportion of the lighter molecules than does the gas remaining behind in the right-hand vessel. Assume as an illustration that the right-hand vessel at first contains an equal number of hydrogen and oxygen molecules. The average velocity of the hydrogen molecules is four times as great as is that of the oxygen molecules, since the weight of a hydrogen molecule is only $\frac{1}{16}$ of the weight of the oxygen molecule. As a result, a given area of the wall will be hit four times as often by a hydrogen molecule as by an oxygen molecule, and four molecules of hydrogen will pass through the opening for every molecule of oxygen which passes through. The left-hand vessel will therefore contain a larger number of hydrogen molecules than of oxygen molecules, after equilibrium is established.

A mixture of gases of different densities may be partially separated by confining it within a porous vessel. The lighter molecules will escape from the vessel through the pores in the walls more quickly than do the heavier molecules, for reasons already explained. For instance, if a quantity of hydrogen and oxygen be confined indefinitely under pressure in a porous vessel, it will be found eventually that the gas in the vessel will consist of four parts by volume of oxygen to one part by volume of hydrogen. The process of separating the constituents of mixtures of gases in this manner is known as osmosis. The same process is employed for concentrating solutions in certain industries.

365. Gases Confined in Conducting Vessels. So far we have been assuming that the walls of the vessel containing the gas have been rigid. This is equivalent to the assumption that they are non-conductors of heat. If a wall is not rigid, the effect of the impact of a molecule upon it will be to cause the wall to vibrate and the molecule will, as a result, surrender energy to the wall. When the amplitude and rate of vibration of the wall is such that the mean energy of the molecules is insufficient to increase the vibration, the walls will have the same temperature as the gas. If the energy (i.e.,

rate and amplitude) of vibration of the wall be increased by some method, e.g., by the application of heat to the surface of the containing vessel, the molecules will, of course, rebound from the walls in the average case with increased energy. This follows from the fact that if a molecule having a given velocity normal to the wall impinges upon the wall while it is retreating with a given velocity, it will lose twice this velocity, while if it impinges upon an advancing wall having the same velocity, it will gain twice this velocity. The gain in energy in the second case is greater than the loss of energy in the first case. Consequently the temperature of a gas is increased by incasing it in walls hotter than the gas itself.

366. The Properties of Imperfect Gases. Thus far, we have considered only those gases in which the molecules are of infinitesmal size and exert no force upon each other when they are not in actual contact. Such gases are perfect gases. In the case of actual gases, the molecules are of finite size. Hence the gas behaves as if it were confined within a volume smaller than the actual volume in which it is confined, by some proportion of the sum total of the volumes of the molecules. We have already seen that the characteristic equation of a perfect gas is $P\,V = W\,R\,T$. We may write then, on the assumption that the molecules in W pounds of the gas have a finite volume whose effective value in diminishing the actual volume of the containing vessel is $W\,b$, the equation

$$P\,(V - W\,b) = W\,R\,T. \qquad (21)$$

It is known that the molecules of actual gases exert a force one upon another. This force is in the nature of an attraction which tends to draw the molecules together and therefore to reduce the pressure which they will exert upon the walls of the containing vessel. Several equations have been proposed in order to show the effect of this inter-molecular attraction upon the behavior of the gas. The best known is that known as **Van der Waals'** equation, which is usually written in the form

$$\left(P + \frac{a}{V^2}\right)(V - b) = R\,T, \qquad (22)$$

in which P is the pressure in pounds per square foot and V is the volume of 1 pound of the gas, T is its absolute temperature, and a and b and R are quantities depending upon the nature of the gas. It is usually assumed in works in physics and physical chemistry that a, b and R are all constants. This is not true, however, since the effect on the inter-molecular attraction upon the behavior of the gas will depend upon the mass of gas considered and upon the form of the containing vessel. The values of a and b will therefore vary according to the circumstances under which the gas is confined.

On this account Van der Waals' equation does not always give concordant results. Clausius has therefore proposed the equation

$$P = R\left(\frac{T}{V - a}\right) - \left(\frac{c}{T\,(V - b)^2}\right), \qquad (23)$$

in which, R, a, b and c are assumed to be constants, depending upon the nature of the gas. As in Van der Waals' equation, however, the values of these quantities depend upon the mass of the gas and the form of the containing vessel, so that the values obtained experimentally will depend upon the circumstances under which the experiment was conducted.

Clausius has shown that when the molecules of a gas attract one another, we may write for the characteristic equation of unit weight of the gas, the equation

$$\tfrac{1}{2} M\,\overline{V}^2 = \frac{3\,P\,V}{2} + \tfrac{1}{2}\,\Sigma\,\Sigma\,R\,r. \qquad (24)$$

In this equation the value of $\frac{1}{2} \Sigma \Sigma R r$ has the following meaning: Let r be the distance between any two molecules of the mass of gas and R the force which they exert upon one another. Each of the molecules exerts a force upon every other one, so that the sums of the products of the several forces each multiplied into the distance through which it is exerted, is represented by the term $\frac{1}{2} \Sigma \Sigma R r$.

It is reasonable to assume that the force which two molecules will exert upon one another is one of attraction and that the amount of this force is inversely proportional to the square of their distance, hence we may write

$$R = \frac{k}{r^2}, \qquad (25)$$

in which R is the force of attraction between two molecules, r is the distance between the molecules, and k is a constant depending upon the nature of the gas. We will have for the term $R\,r$, the value $\frac{k}{r}$ and for the value of the term $\frac{1}{2} \Sigma \Sigma R r$ we may substitute the expression $\frac{K}{r}$ and write the equation in the form

$$P\,V = \tfrac{1}{3} M \overline{V}^2 - \frac{K}{r}, \qquad (26)$$

for 1 pound of the gas. Now the value of the quantity $\frac{1}{2} \Sigma \Sigma R r$ will be proportional to the square of the number of molecules, so that we may write for W pounds of the gas, the expression

$$P\,V = \tfrac{1}{3} M \overline{V}^2 - \frac{K W^2}{r}. \qquad (27)$$

In this equation, r is proportional to the distance between any pair of molecules and is, therefore, proportional to the cube root of the volume in which the gas is confined. Writing this so and taking account of the effect of the volume of the molecules, we will have finally for an expression giving the relation between the pressure, volume, and temperature of an imperfect gas

$$P\,(V - W\,b) = W\,R\,T - \frac{a\,W^2}{\sqrt[3]{V}}, \qquad (28)$$

in which b and r are constants for any given gas, and a is a quantity which depends upon the form of the containing vessel, and the nature of the gas, but is independent of the mass of the gas.

In general experiments upon gases are conducted in cylinders of unchanging diameter but of changeable length, so that the form of the containing vessel is continually changing during the progress of the experiment. It cannot therefore be expected that any rational equation can be derived for the behavior of a mass of gas in such a vessel.

367. The Phenomena of Liquefaction. When an imperfect gas is sufficiently cooled and compressed, the slower moving molecules in any region of small extent will be drawn together by the attractive forces, and on account of their low kinetic energy will be unable to separate themselves, but will continue to travel about the common center of attraction in an irregular orbit of some form. Wherever several molecules are drawn together in this manner, a point is produced at which attractive forces are centered, so that other molecules will be attracted toward that point. Those having high velocities will pass the point, while those having low velocities will be drawn into the system and help to increase its attractive force. When this condition occurs, a gas begins to condense into a liquid. The action will be accelerated by the presence of small particles of dust which tend by their attraction to gather about

themselves quantities of molecules. As fast as such systems of molecules are formed they are drawn together by their attractive forces so that finally the whole mass will condense into a liquid provided that the pressure and temperature are kept constant. In approaching these centers of attraction the velocities of the molecules are increased by the attractive forces, which, of course, increase the temperature of the mass. This increase in temperature must be counteracted by the withdrawal of heat. The energy developed by the movement of the molecules under these attractive forces is the latent heat of evaporation of the liquid.

Since all of the molecules do not have the same velocity, and the distribution of velocities among the molecules is determined by the laws of chance, it follows that the pressure at which condensation occurs, for a given temperature, does not necessarily correspond to any particular point upon the isothermal of the vapor, but is determined by the distribution of the velocities. It may therefore be shown that the relation between the pressure and temperature of vaporization of any vapor may be expressed by an equation of the form

$$\log_e P = f(T). \qquad (29)$$

It may also be pointed out that the temperature of condensation for a given pressure is independent of the mass of the vapor considered and of the form of the containing vessel, since this condensation is due to the concentration of molecules at local points and is not due to the simultaneous concentration of the whole mass of vapor by the sum total of all the attractive forces.

368. The Phenomena of Vaporization. The molecules of a liquid may be assumed to be in constant agitation in the same way as are those of a gas. In the case of a liquid, however, the molecules are restrained from leaving the mass freely on account of their attraction for one another, If the surface of a liquid be assumed to be in contact with its own vapor, the surface of the liquid will be continually struck by molecules of the vapor. In order that the liquid shall not continually gain in mass, it is necessary to assume that when the the liquid and vapor are in thermal equilibrium (i.e., when they have the same temperature) the number of molecules leaving a given area of the liquid in a given time is equal to the number of molecules of the vapor striking this surface in the same time. The molecules which leave the liquid are necessarily those which have a superior velocity. Consequently, when a liquid suffers the loss of some of its molecules by being in contact with a mass of its vapor whose saturation temperature is lower than the temperature of the liquid, those particles which leave the liquid (i.e., which are evaporated), are those having superior velocity. Consequently, by the evaporation of some of its particles, the temperature of the liquid will be lowered. In escaping from the liquid, the particles lose that portion of their kinetic energy which is expended in separating them from the remainder of the mass of liquid, against the attractions of the molecules of the liquid.

In the case of a sensibly perfect gas, the number of molecules which strike a given surface in a given time is proportional to the number of molecules per unit of volume and also to the velocity of the molecules. From the characteristic equation of gases we may write

$$\frac{W}{V} = \frac{P}{RT}, \qquad (30)$$

in which $\frac{W}{V}$ is proportional to the number of molecules per unit of volume. The mean velocity of the molecules is proportional to the square root of the absolute temperature, so that the molecules striking upon a given surface in a given time, in the case of a sensibly perfect gas, is proportional to the pressure of the gas divided by

the square root of its absolute temperature. In the case of a vapor, the number of molecules striking a given area of the surface of a liquid in a given time is nearly proportional to the pressure of the vapor, so that the rate of evaporation from the surface of a liquid is nearly proportional to the vapor pressure corresponding to the temperature of the liquid. Hence, when the surface of a liquid is in contact with a gaseous medium in which the pressure of its vapor is less than the saturation pressure corresponding to the temperature of the liquid (as, for instance, when a body of warm water is exposed to cold air), the rate of evaporation from the surface of the liquid is nearly proportional to the difference between the pressure of the vapor and the vapor pressure corresponding to the temperature of the liquid.

369. Diffusion of Gases. On account of the attraction which the molecules of a gas have for one another, they do not pass from point to point in straight lines, but through the influence of the neighboring molecules, the directions of their paths are being continually changed. On this account, a molecule, although it has a high velocity, is very slow in moving from point to point.

If two gases be separated from one another by an imaginary surface, the molecules of one gas will tend to pass between those of the other gas and the two will mix, the process being known as diffusion. However, the progress of the molecules is slow on account of the inter-molecular attractions, so that diffusion is a gradual and comparatively slow process. The higher the velocity of the molecules, the faster will be the rate at which they diffuse into one another, so that at high temperatures, gases diffuse faster than at low temperatures and light gases diffuse faster than do heavy ones. On the other hand diffusion is retarded by high pressures.

Vapors diffuse through gases in the same way as do other gases. Consequently, if a gas be in contact with a liquid, the evaporation of the liquid will cause its vapor to diffuse into the gas. The pressure of the vapor in contact with the liquid will be very nearly the pressure of the gas, in case the temperature of the liquid is such that the saturation pressure of its vapor is equal to or greater than that of the gas. In case it is less than the pressure of the gas, the pressure of the vapor in contact with the liquid will be very nearly the saturation pressure corresponding to the temperature of the gas. By the continual diffusion of the vapor into the gas, the pressure of the vapor in contact with the liquid will become a trifle lower than the saturation pressure at the temperature of the gas.

This accounts for the phenomena of evaporation in air which is not saturated with moisture. If a surface of water be exposed to air of the same temperature, the water will evaporate and the air in contact with the water will become saturated with water vapor. This water vapor will diffuse away, lowering the vapor pressure sufficiently to permit of more evaporation. The temperature of the water will fall, on account of this evaporation, until finally equilibrium is established, and the rate at which the water receives heat by radiation and conduction from surrounding objects will become equal to the rate at which it parts with heat by evaporation. The pressure of the water vapor in the air in contact with the water will then be slightly lower than the saturation pressure corresponding to the temperature of the water. The rate of evaporation will then depend upon the humidity of the air and the rate of diffusion of the water vapor, in case the air is still. In case there is a wind, the process is greatly accelerated.

370. Dissociation of Gases. It has already been pointed out that the atoms which constitute a molecule have motion relative to one another and that, in consequence, the gas has intra-molecular energy. As the temperature of the gas is increased, its intra-molecular energy is also increased. It is reasonable to assume that the internal energy of a molecule is due to the rotation of its several atoms about their common center of attraction, exactly as the bodies of the solar system revolve about their

common center of attraction, under the influence of gravitation. So long as such a system of atoms is unacted upon by an external force the sum of the potential and kinetic energies of the system remains constant and the system remains a unit. If some external force, (e.g., a collision with another molecule), increases the sum of the potential and kinetic energy of the system to some value greater than the potential energy which the system would have were its several atoms removed to an infinite distance from one another, the system is permanently broken up and thereafter consists of two or more separate atoms. If any of these atoms then encounters another atom upon which it can exert an attraction, the two would combine to form a new system, provided their initial relative velocity is not such as to prevent.

In a gas, the internal energy of most of the molecules is very nearly the mean of that of all the molecules. A few of the molecules, however, will have internal energies greater in excess of the mean value and the energies of some of these will be so great that the molecules will be disintegrated. As the temperature of the gas is raised, the number of molecules so broken up will be increased until finally a sufficient number of them are so separated that their presence has an appreciable effect upon the properties of the gas. This phenomenon is termed dissociation.

If two gases capable of forming a chemical compound are mixed together, they will not usually react with one another. Occasionally, however, a molecule of one gas will be broken up and the separate atoms, meeting separate atoms of the other gas, combine with them to form a molecule of the compound. The formation of the compound at ordinary temperature progresses with very great slowness, since the number of dissociated molecules is very small. If the temperature be sufficiently raised, however, so that an appreciable number of the molecules of either of the gases are dissociated, the reaction will become more rapid. When it becomes so rapid that heat is generated by the reaction at a faster rate than it is radiated, the kindling point of the mass is reached, since the increase in temperature then accelerates the reaction, and the reaction in turn accelerates the increase in temperature, and the reaction is thereafter self-sustaining.

The generation of heat by the reaction will raise the temperature of the whole mass of gas. As a result of this increase in temperature, some of the compound will be dissociated. At the temperature which results from the reaction a very considerable proportion of the compound is thus dissociated, so that the full amount of the heat of combustion is not immediately developed by the reaction. This is the phenomenon spoken of in Chapter XX as delayed combustion. As the gas cools by radiation and conduction, the number of molecules containing more than the limiting quantity of internal energy will be reduced, and the remainder of the heat of combustion will be evolved, until at ordinary temperatures practically none of the molecules of the compound are dissociated.

371. Variation of the Specific Heats of Gases. The effect of the dissociation of a gas at high temperatures is, of course, to change the nature of the gas. It also changes the apparent specific heat of the gas by increasing it. Since the number of molecules is increased (on account of the separate atoms becoming molecules) the pressure and volume of the gas at a given temperature is also increased.

At low temperatures a gas confined at constant volume has a constant specific heat. If, however, the gas be confined at constant pressure, the specific heat will be found to vary, since, as the volume diminishes, a portion of the potential energy which resides in the gas in virtue of the attraction of its molecules is given up in the form heat. As the volume of the gas diminishes, the strength of the attractive forces increases, and the amount of potential energy transformed into heat by a given temperature reduction also increases. Consequently, in the case of an imperfect polytomic gas, the specific heat at constant volume increases gradually as its temperature is raised,

and the specific heat at constant pressure at first decreases and then afterward increases as the temperature is raised. The variation in the specific heat at constant pressure of permanent gases having a high dissociation temperature will, however, be so small at ordinary temperatures as to defy measurement. In the case of monatomic gases, there will be no variation in the specific heat at high temperatures, since there can be no dissociation of such gases. It follows that a constant pressure thermometer using a monatomic gas as its thermometric fluid, will give results which agree perfectly with the thermodynamic scale of absolute temperatures, except for the accidental errors of the instrument itself. It is to be regretted, therefore, that hydrogen and not helium was chosen as the thermometric fluid in the standard thermometer, although at ordinary temperatures, it is quite probable that the indications of the hydrogen thermometer are so exact that the errors resulting from dissociation are completely masked by the accidental errors of the instrument itself.

INDEX